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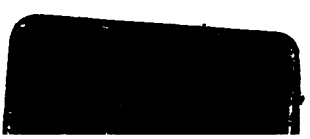

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MOTOR VEHICLE ENGINEERING

THE CHASSIS

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MOTOR VEHICLE ENGINEERING

THE CHASSIS

BY

ETHELBERT FAVARY

MEMBER SOCIETY OF AUTOMOTIVE ENGINEERS, CONSULTING ENGINEER, AUTHOR OF "MOTOR VEHICLE ENGINEERING-ENGINES"

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PREFACE

In this volume, "The Chassis," as in the first of this series, "Engines," the author endeavors to present in simple language and with only elementary mathematics, a textbook containing information required by the automotive engineer and the designer. He hopes it will be found useful also in engineering offices, and in institutions where the subject of Motor Vehicle Engineering is taught.

The book is not a record of the development which led to the present state of the art, nor is its purpose merely to make known present day practice as exemplified in the motor vehicles now manufactured, but to impart the underlying theory that students, and those engaged, or wishing to become engaged, in the automotive industry, may be enabled to apply the information herein contained in actual work.

In some instances, facts and examples are repeated in different forms as the author has found from his experience in imparting technical knowledge to persons who have not had sufficient technical preparation, that, often, by varying explanations, many are enabled to grasp the problems who otherwise could not follow them.

The method pursued in the treatment of the subject is largely a result of the author's work as consulting engineer, his experience with draftsmen and designers and his lectures on Motor Vehicle Design at Cooper Union to men engaged in the industry.

Many examples are devoted to the commercial vehicle, which is at present an especially timely topic.

The author is indebted to the large number of manufacturers and to the War Department who have supplied him with material in the shape of data and drawings, to the Society of Automotive Engineers for permission to use extracts from its various publications, and to George F. Thackery, for the tables on pages 46 to 49 from the Cambria Steel Handbook. It is only through this coöperation that the author has been enabled to present this volume in such comprehensive form.

When reference is made to Volume 1 in the text, it denotes the first book of this series, "Engines."

The author would be grateful for suggestions for the improvement of future editions of this book and for drawing his attention to any errors which may have been overlooked.

ETHELBERT FAVARY

NEW YORK, N. Y.,
August, 1922.

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MOTOR VEHICLE ENGINEERING

THE CHASSIS

CHAPTER I

CHASSIS LAYOUT

The chassis of an automobile is the frame structure and the entire mechanism supported by it, including the engine and the wheels but not including the body. The engine was considered in the first volume of this series entitled "Motor Vehicle Engineering, Engines" which will hereafter be referred to as Vol. I. In the present volume are treated the various portions or units forming the chassis, exclusive of engine details.

The engine is placed near the front end of the chassis and is supported by the frame direct, though, occasionally to avoid the severe loda on the frame side members, it is supported by a sub-frame. The latter, however, adds weight and cost and is therefore as a rule not employed.

The latest practice in motor vehicle construction is to combine the motor, clutch and transmission, into a single unit and this is termed the "unit power plant." In a number of machines, more especially in truck construction, the transmission is placed "amidship," that is to say, it is contained in a separate unit placed about the middle of the chassis, behind the clutch. Occasionally (though rarely), the transmission is placed at the rear of the propeller shaft in a unit with the rear axle, or in a unit with the torque tube. The motor is supported by three or four supports, the former is called the "three-point suspension," and by its use the bearings are considerably relieved of strain when the frame is subjected to severe stresses, as for instance, when the vehicle is surmounting obstructions on the road or when rapidly turning corners. In such event, the strain (twisting or bending) in the frame tends to throw the shaft out of alignment; the

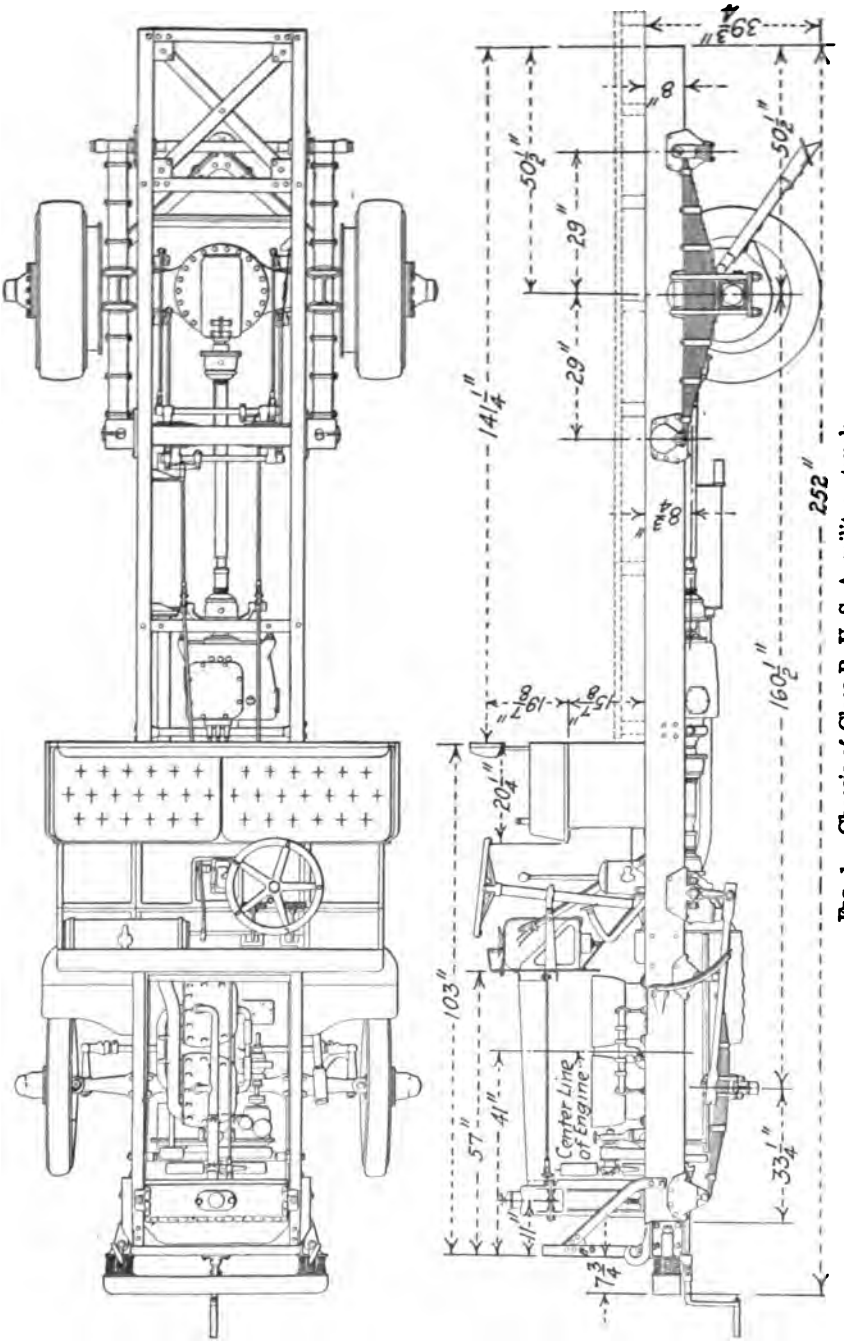


FIG. 1.—Chassis of Class B, U. S. A. military truck.

"three-point suspension" readily accommodates itself to changes in the lines of the frame without transmitting them to the units suspended on it. At times, some of the supports of the units are designed to give them a certain swivel action in relation to car frame.

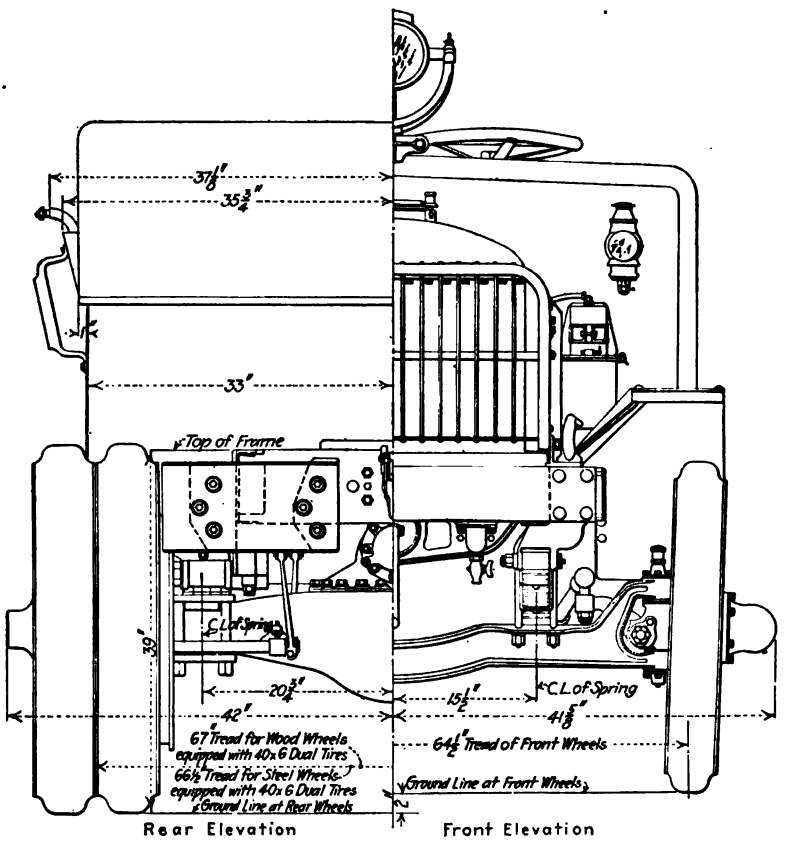


FIG. 1a.—Class B truck.

One of the problems in chassis design is to have the center of gravity as low as possible (see Chap. on Springs) as this contributes to ease of riding. Furthermore, a low frame renders possible a low body and this facilitates stepping into and out of the automobile. When the frame is set low, the angle between the propeller shaft and the motor shaft is smaller, that is to say, they are in better alignment (if the motor shaft runs

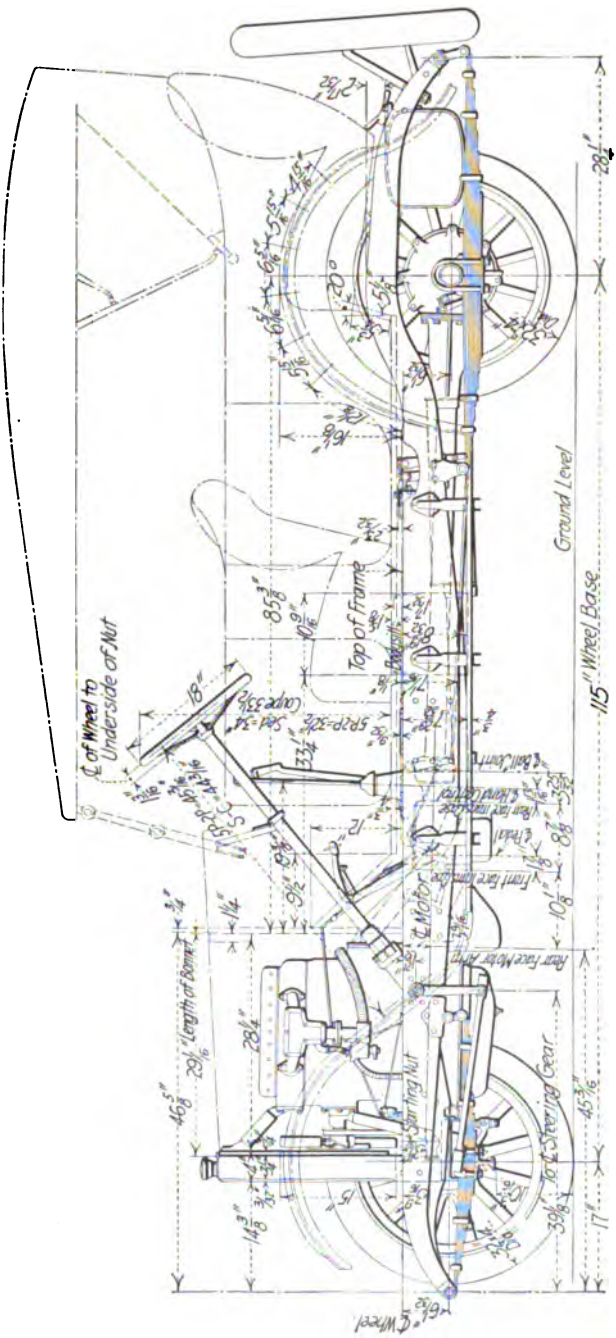


FIG. 2.—Center-line layout of 4-cylinder Oldsmobile. Side view.

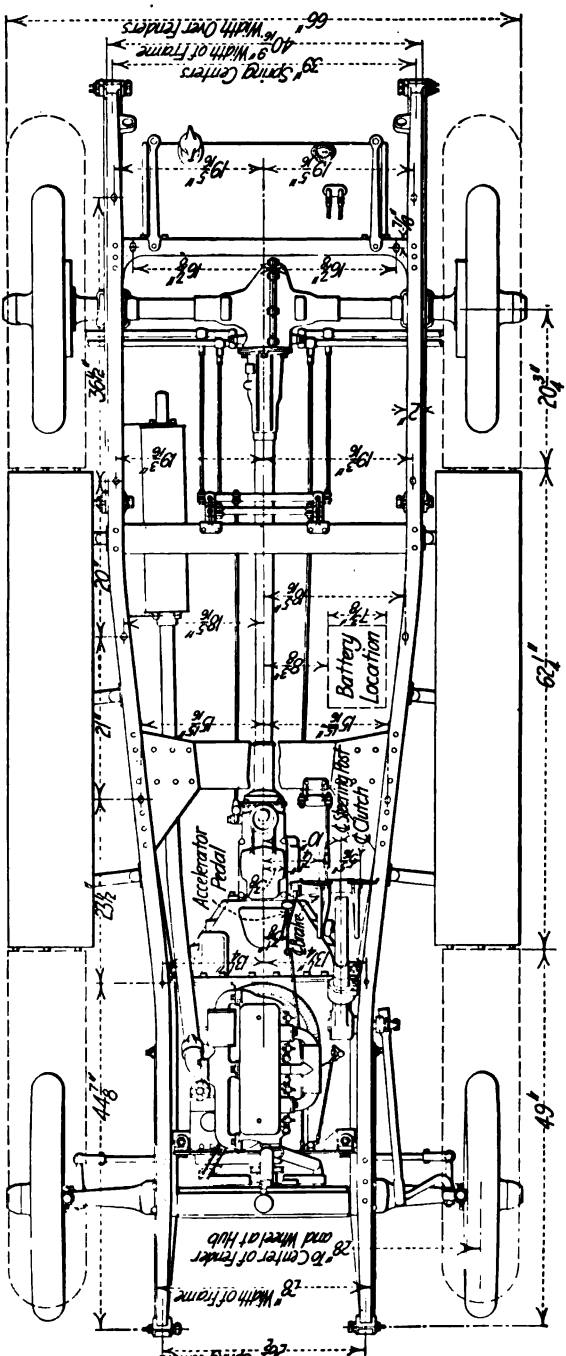


Fig. 2a.—Center-line layout of Oldsmobile. Plan view.

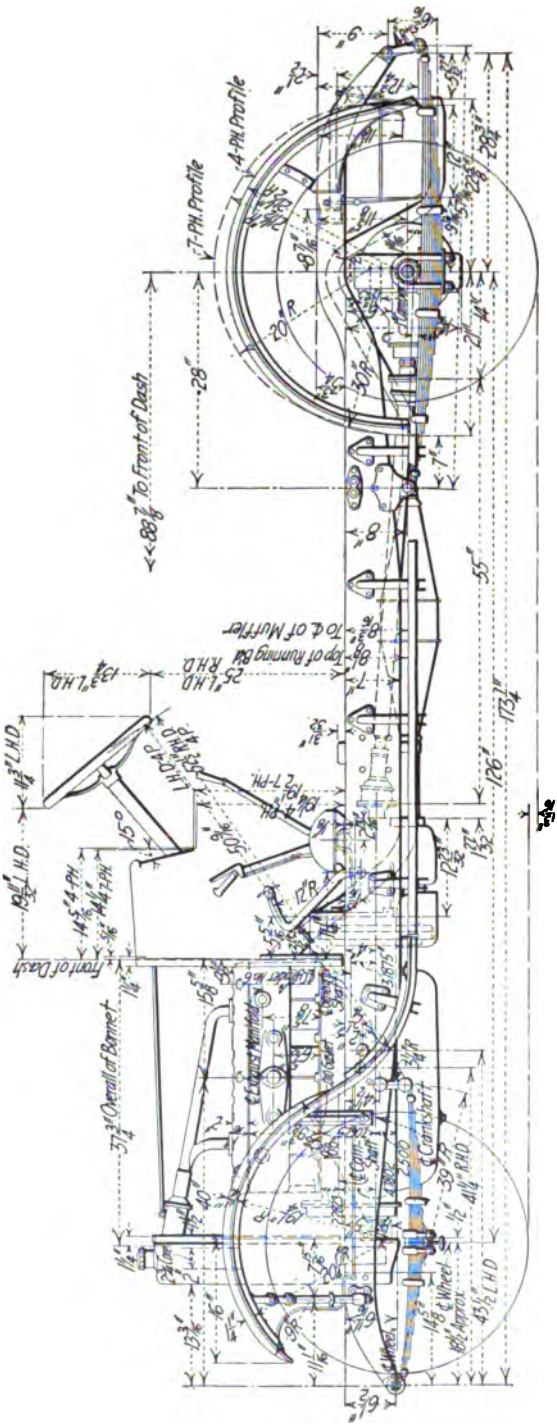


FIG. 3.—Center-line layout of six-cylinder Hudson. Side view.

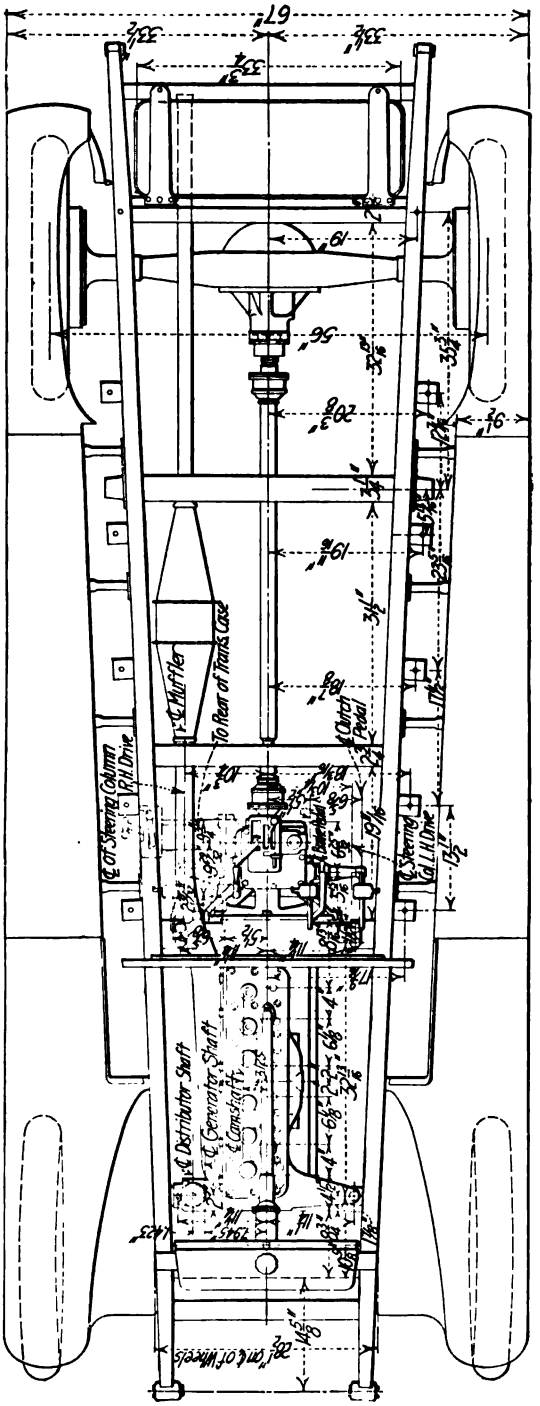


FIG. 3a.—Center-line layout of six-cylinder Hudson. Plan view.

parallel to the frame). Sometimes the motor supporting arms are made rigid and take the place of cross members by interconnecting the two side members, to stiffen the construction. In the front of the frame there is about 11 in. clearance between the side of the frame and the inner side of the wheel or tire, to allow sufficient space for the swing of the front wheel, when turning a corner. Figure 1 shows the chassis layout with the principal dimensions, of the "worm driven" U. S. A., Class B, Military Truck, which has a four-cylinder engine with a

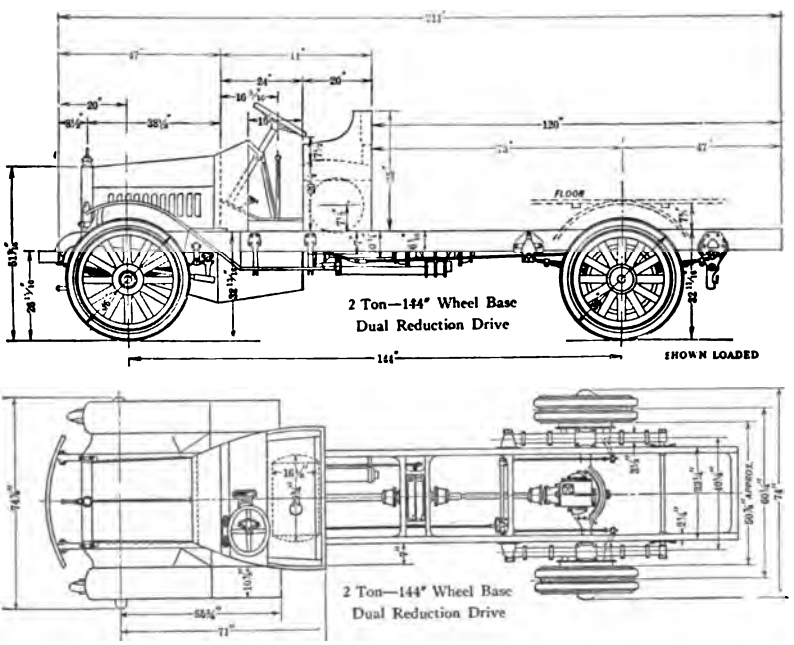


FIG. 4.—Mack 1½- and 2-ton double reduction truck.

bore and stroke of 4¾ by 6 in. The weight of the chassis is about 8,000 lb., and the weight complete with full load 17,000 lb. Figure 1*a* shows the front and rear elevations of this chassis.

Figures 2 and 2*a*, give the center line layout (side view and plan view) of the four-cylinder Oldsmobile touring car (Model 43A) having a bore and stroke of 3 1/16 by 5¼ in. Its weight, empty, is about 2,900 lb.

Figure 3 is the center line layout of the six-cylinder Hudson touring car (bore and stroke 3½ by 5 in.) weighing empty,

3,400 lb. and 3,600 lb. for the 4-passenger and 7-passenger models respectively.

Figure 4 shows the general dimensions of the 1½-and 2-ton

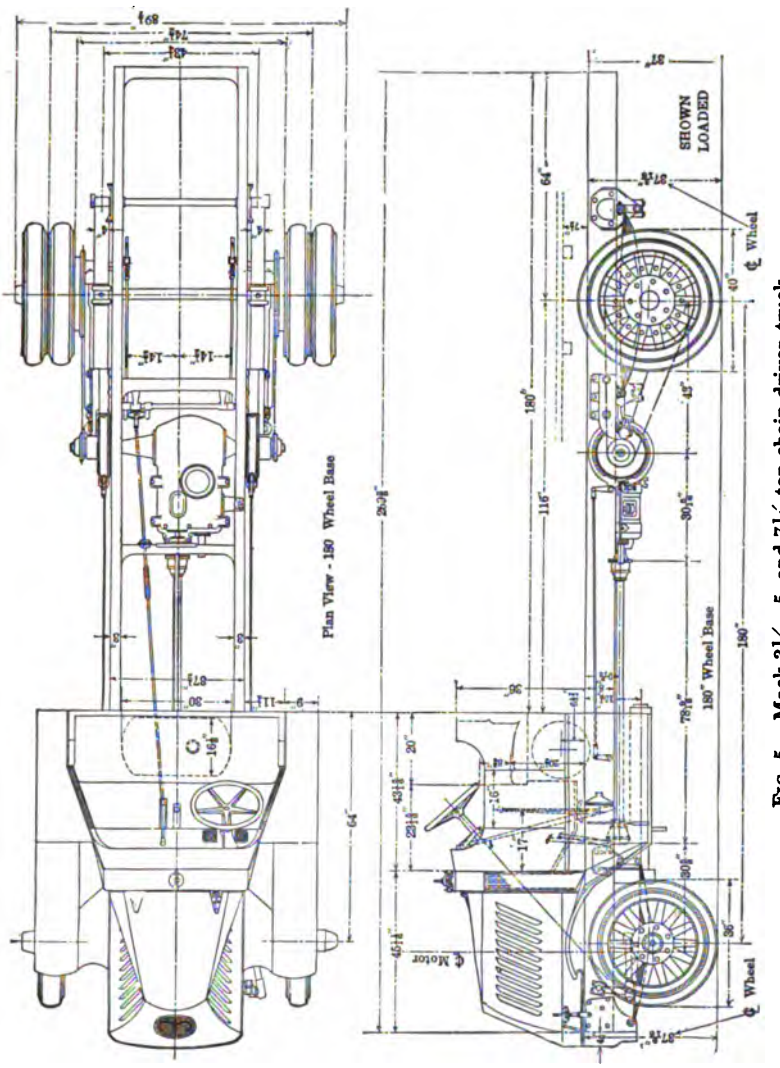


FIG. 5.—Mack 3½-, 5- and 7½-ton chain driven truck.

"Double Reduction" Mack truck. The frame is of channel section 6 3/16 in. high, 2¼ in. wide and made of ¼-in. stock. This truck is furnished with a unit power plant and has three universal joints between the rear axle and the motor.

Figure 5 gives the general dimensions of the 3½-, 5- and 7½-ton Mack chain driven trucks. The frames of all three consists of 8 by 3 by ¼-in. channels; in the first two sizes the material is flexible pressed steel, the 7½-ton truck has a pressed chrome nickel steel frame. The difference in the three sizes consists of heavier tires for the larger capacities as well as lower gear ratios; this will be discussed later in the chapter on "Rear Axles." In these chain driven trucks the transmission forms a unit with the counter shaft (often called jackshaft) and the differential.

Before treating of the design of frames the strength of materials entering into their construction, will be discussed. The side members and cross members of a frame are considered beams. See also Chap. XXVII of Vol. I, "Materials," "Modulus of Elasticity," "S.A.E. Steels," etc.

CHAPTER II

MECHANICS OF MATERIALS

Strength of Materials, Beams.—The ultimate strength of a material is the unit load which will cause rupture or fracture of a test piece of a given (original) cross-section. When the material is in tension, the ultimate strength is called the tensile strength, and when in compression, the compressive strength. If a test piece has originally (before the test), a cross-section of A sq. in. and if fracture occurs when loaded with W lb., if T is the ultimate strength of the material tested per unit area, that is to say, per square inch, then $T = \frac{W}{A}$ lb. per square inch. Experiments conducted by various investigators, have disclosed the fact that materials have a lower ultimate strength after they have been subjected to loads of varying intensity, instead of a steady load. Their strength is lower still, when the load varies and alternates in direction, *i.e.*, when it is in tension and compression at intervals; loads suddenly applied, or shocks, will reduce the ultimate strength still further (*i.e.*, if the material has been subjected to a number of such loads or shocks).

If the ultimate strength of the material under tension or compression is designated by T when subjected to a steady or dead load, its ultimate strength when subjected a few million times to variable loads will only be about $\frac{2}{3}\ T$, and if the load is reversed so as to cause tensile and compressive stresses alternately, its ultimate strength will finally be only approximately one-third of its original strength T. This fact should be considered when determining the necessary factor of safety, which will be explained later. Test pieces for testing the tensile strength of a metal are usually made 2 in. long.

Elongation.—"Elongation" is the term given to the increase in length in the material just before rupture takes place. For example, a test piece originally 2 in. long, is $2\frac{1}{2}$ in. just before rupture; the elongation in 2 in. is $\frac{\frac{1}{2}}{2} = .25$ or 25 per cent.

Reduction in Area.—Whenever a specimen is tested for tensile strength it experiences a "reduction in area." If A is the area of the section before the test, and a the final area at the place of rupture, the reduction in area $= \frac{A-a}{A}$, and by multiplying the result by 100, we obtain the answer in per cent.

Factor of Safety.—The factor of safety f of a material is the ratio of the ultimate strength to the permissible working stress, *i.e.*, its ultimate strength T (in pounds per square inch) divided by the working load w, in pounds per square inch. Thus

$$f = \frac{T}{w}, \text{ and } w = \frac{T}{f}$$

For instance, if the ultimate strength of a steel bar is 80,000 lb. per square inch, and the working load 16,000 lb. per square inch, then the factor of safety $f = \frac{80,000}{16,000} = 5$. The following are the factors of safety which the author finds satisfactory in practice.

TABLE I.—FACTORS OF SAFETY

Material	Dead load	Variable load in one direction	Variable load in tension and compression	Shocks or loads suddenly applied
Aluminum	5	8	12	18
Cast iron	5	8	12	18
Copper	5	8	12	18
Wrought iron	4	6	8	15
Steel	4	6	8	15
Timber	6	9	16	20

The ultimate strength of steel is approximately the same when the material is in tension or compression. For other metals, like cast iron, it varies considerably, being very much weaker in tension. Hence, the factor of safety should be used with respect to its tensile strength when the material is only in tension; if it is in compression or under a shearing stress, a factor of safety should be provided which will take care of these the stresses. All stresses in a structure or a bar may be resolved into tensile, compressive and shearing stresses, and the strength of the material

to resist such stresses are termed the tensile, compressive and shearing strength respectively. Bending stresses are chiefly a combination of tensile and compressive stresses and also to some extent shearing stresses. Torsion stresses, like the twisting of a bar for instance, are in reality shearing stresses.

TABLE II.—STRENGTH OF MATERIALS

Material	Tensile strength in pounds per square inch	Compression strength in pounds per square inch	Shearing strength in pounds per square inch	Modulus[1] of elasticity (when in tension)
Aluminum (cast)	18,000	14,000	14,000	11,000,000
Aluminum alloy (5 per cent copper)	26,000	44,000	23,000	
Brass (cast)	24,000	30,000	35,000	9,000,000
Bronze gun metal	30,000	60,000	22,000	11,000,000
Bronze manganese	60,000	110,000	90,000	12,000,000
Iron (cast)	10,000–30,000	75,000–100,000	.....	15,000,000
Iron (wrought)	50,000	45,000	40,000	25,000,000
Steel (casting)	70,000	70,000	60,000	30,000,000
Steel	60–150,000	Same as in tension approximately	15 per cent less than in tension approximately	30,000,000

[1] See Chap. XXVII, Modulus of Elasticity, vol. I.

Yield Point.—In Chap. XVII. Vol. I it was shown that the elastic limit of a material is reached when the application of a load causes a permanent set. With some materials, the elastic limit is very low, for a small load (small, compared to its ultimate strength) will cause a permanent set, but when the elastic limit is exceeded only by a small extra load, the permanent set is very small, and only when increasing the load considerably beyond the elastic limit will the permanent set become large. In like manner the strain in the metal *i.e.*, the deformation under load, will up to the point of the elastic limit of the material be very small and will increase thereafter at first only very gradually with an increase in load up to a certain limit. Only after the elastic limit has been exceeded considerably, will the strain, (the deformation in the material) increase rapidly. The point

where the deformation first begins to increase rapidly without an increase in load (which is also the point where the permanent set begins to become large) is called the yield point of the material; this term is often used, for with certain materials it is a better guide as to the maximum load which the material will safely carry. In steel or other metals, the yield point and the elastic limit are not very far apart, but occasionally even with steel, the yield point is noted instead of the elastic limit.

Figure 6 shows the stress and strain curve of soft steel, and the foregoing may be summarized as follows:[1]

1. Section area—as applied to a bar under tensile or compressive stress is that area of the unstressed bar measured in a plane at right angles to the direction of said stress.

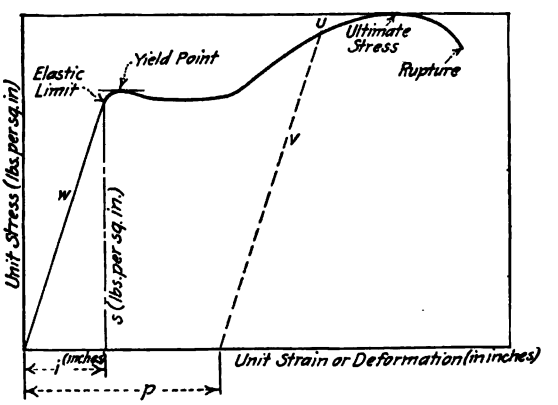


FIG. 6.—Stress and strain curve of soft steel.[1]

2. Stress—is an internal induced force which tends to resist deformation by external applied force.

3. Strain—the deformation produced by the external applied force. (When the material is in tension, the strain is the increase in length.)

4. Elasticity—that property by virtue of which a material tends to resume its original form after removal of the external applied force.

5. Elastic limit—that unit stress for a given material beyond which the material will not resume its original form after removal of an external force.

6. Yield point—that unit stress for ductile materials beyond

[1] From Cooper Union Course in Strength of Materials.

which the deformation continues without increase or with decrease in applied force.

7. Ultimate strength—the highest unit stress obtainable in a material when tested to complete rupture based on the *original* cross sectional area of the test specimen.

8. Modulus of elasticity—$\frac{S}{i}$ *viz.*, the ratio of the unit stress to the unit strain within the elastic limit of the material.

9. Permanent set—the difference between the original form of the test specimen and the final form after the force has been removed which has stressed the specimen beyond the elastic limit. If for instance, the load were removed when the point u is reached, (Fig. 6), the deformation will decrease and follow line v as the load is diminished, (this line is parallel to w) and when the entire stress in the material is removed, the permanent set is p (in inches).

If a weight is suspended on a vertical steel bar of an area of A sq. in. (before the load is applied) and if the load is W lb., then the tension T in the bar per square inch is $\frac{W}{A}$. If W is increased until the bar ruptures, then the ultimate strength or the tensile strength T, (in pounds per square inch) is $\frac{W}{A}$. Likewise when the material is compressed, its compressive stress $C = \frac{W}{A}$, lb. per square inch. Figure 6*a* illustrates what is meant by shearing stress. "*a*" represents a bar of steel, and *c* and *d* are tools moving in the direction of the arrows. The tools are, so to say, cutting the material along the dotted line and this action is termed "shearing" (which is the same when punching a hole in sheet metal). If A is the sectional area which is actually sheared by a force or load W, and if S stands for the shearing strength of the material per square inch, $S = \frac{W}{A}$. When shearing a bar, the area is simply the cross-section of the bar. When punching a hole of diameter d, in sheet metal of thickness t, the

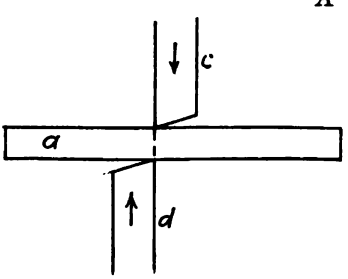


FIG. 6*a*.—Illustrating shearing stress.

area of metal sheared is πdt, and the load or force W required to punch a hole is $S\pi dt$. However, shearing stresses will also take place when the tools (they may simply be supports) are far apart horizontally. In such event, the shearing may be considered to take place all along the beam, as shown by the vertical dotted lines in Fig. 8.

Bending Stresses.—When a bar or a beam is loaded with a weight W in such a manner as to bend it, the metal on one side of the beam is under tension and on the other side under compression. In frames of motor vehicles the beams are considered as "free" at the ends (they are partly restrained) and not restrained like beams in a wall.

Simple Beam.—Consider first what is called a simple beam, (see Fig. 7), which is resting on supports H-H near its outer ends and is loaded between the supports by weight W. (Any horizontal

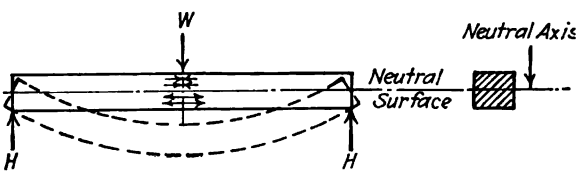


FIG. 7.—Simple beam.

structure which is supported at one or more points and supports one or more loads may be termed a beam.) On the top the fibers will be in compression as shown by the arrows, while on the bottom they will be under tension, hence the beam will tend to bend downward, as shown dotted. Evidently there is a line through the center of gravity where the material is neither under tension nor compression; this line (imagined to run through the entire beam and exposing a surface) is called the neutral surface. The line projected by the neutral surface through a cross-section of the beam is called the neutral axis.

Simple Cantilever Beam.—Figure 8 shows what is called a simple cantilever beam. In this case the application of a load W will cause a compression, (shown dotted) in the material at the lower side, and a tension (shown in thin dashes) on the upper side.

Moment of a Force.—The moment of a force with respect to a point is the magnitude of the force multiplied by the distance from the given point to the line of action of the force, (the distance being at right angles to said line of action) (see Fig. 9).

Suppose a force of 15 lb. exerted at the end of a lever, 8 in. long, in a direction at right angles to the lever, and the lever attached to a shaft; the moment of the force tending to rotate the shaft to which the lever is attached is $15 \times 8 = 120$ lb.-inches. On the other hand, if the line of action (or the direction) of the force is not at right angles to the lever, but as shown in Fig. 10, the distance will not be 8 in., but it will be the length of a line, in this

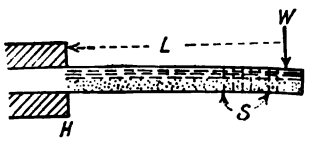


FIG. 8.—Simple cantilever beam.

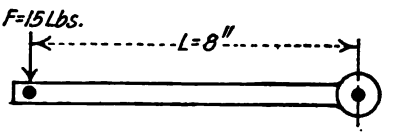


FIG. 9.

case 3 in., from the given point O to the line of action, and at right angles thereto. The moment of the force is therefore $3 \times 15 = 45$ lb.-inches. *The moment* may be said to be a measure of the tendency of the force to produce rotation around a given point. When the moment of the force tends to produce a clockwise rotation about a given point it is said to be positive, and if the tendency is counter-clockwise, it is negative.

Bending Moment.—The bending moment or the force tending to bend the beam at H, (Fig. 8), will depend on the magnitude of the force or load W, in pounds, and on the distance, L, of the load, from the fixed support H. If L is the length in feet, the bending moment at H, which is in this case the maximum bending moment $B = WL$, and is expressed in pound inches or pound feet, depending as to whether L is specified in inches or feet.

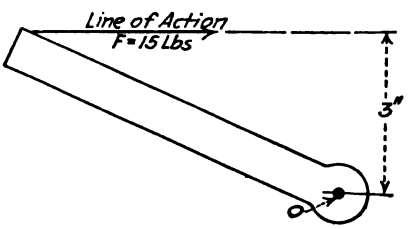


FIG. 10.

Many writers and men of science use the term "foot-pounds" or "inch-pounds," when discussing moments of forces. Others on the other hand, use "pound-feet" or "pound-inches." Inasmuch as in all English speaking countries the term "foot-pounds" or "inch-pounds" are used to denote work or energy, the author will employ the terms "*pound-feet*" and "*pound-inches*," when discussing forces or moments of forces, (the same as is generally used for expressing torque, see Chap. XX Vol. I) to distinguish such forces from work or energy.

The bending moment is the moment tending to bend the beam; in this case it is the moment tending to rotate the beam about point H. The reaction at H, *i.e.*, the force holding it up will equal WL, (as above) since *action and reaction are always equal* to each other but opposed in direction. There will also be a tendency for the beam to be sheared along its entire length (in the direction shown by dotted lines S) and this shearing force is equal to the load W. When a beam or a body is at rest, the algebraic sum of all the external forces acting on it, equals zero, since it is in a state of equilibrium.

Figure 11 illustrates graphically that the strength of a beam depends principally on its height and on the amount of metal in the upper and lower flanges. The bending moment B, at x, (the force tending to rotate the beam around point x) is the force or the load (in pounds) W, (acting vertically downward) multiplied by the horizontal length L (in inches or in feet) from x to the application of the load.

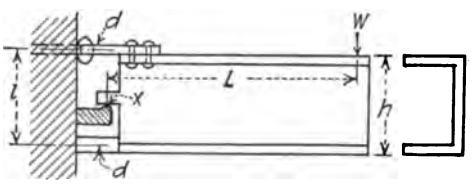


FIG. 11.—Illustrating strength of beam.

Bending moment $B = WL$ lb.-inches or lb.-feet. (We are neglecting the weight of the beam in this case.) Evidently the force F holding the beam up must be equal to the force tending to bend it down, that is to say $F = WL$. The force F is the length l multiplied by either the tension on the top or the compression at the bottom in supports "d." The tension in the block "d" on the top is equal to the compression in the block at the bottom, while in the center the beam is neither in tension or compression. If we call S the total stress caused by either the tension or compression, we have $lS = WL$; hence $S = \frac{WL}{l}$; this formula shows that the greater the distance l between the blocks, the smaller will be S for a given value of L and W, hence the stronger is the beam. For let $L = 10$ ft. and the weight W 50 lb., we have the maximum bending moment B tending to rotate the beam around point x) $= WL$, and $WL = lS$, therefore $S = \frac{WL}{l} = \frac{50 \times 10}{l}$; if l is 2 ft., then $S = 500/2 = 250$ lb. = the tension or the compression (they are equal to each other) in the small

blocks "d." If l is greater, the stress is smaller (since $lS = 500$) hence, the higher the beam or the greater the distance l between the upper and lower blocks (of a given area) the greater is the load which the beam can support.

If a scale were placed at x it would be found that the force or weight tending to let the beam drop straight down vertically

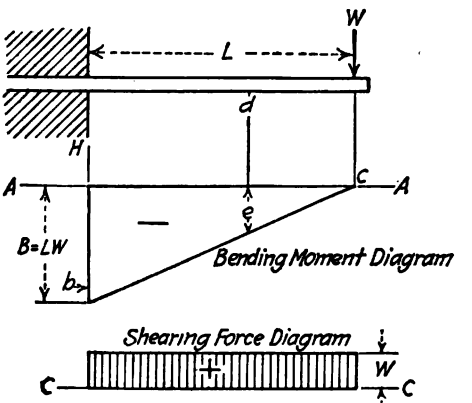


FIG. 12.—Cantilever beam stresses.

is equal to the total weight of the beam and the load W, which is the shearing force or if we neglect the weight of the beam for the present, the shearing force, $= W$. When laying out beams it is often desired to find the magnitude of the bending moment at every point of the beam, and to this end a bending moment diagram is drawn, (see Fig. 12). When the load tends to bend the beam concave upward, (see Fig. 13) the bending moment (for the sake of drawing the bending moment diagram) is termed

FIG. 13.

FIG. 14.

positive (+) and when it tends to bend it with the concave side downward (Fig. 14) it is termed negative (−). Suppose W (Fig. 12) is a load of 50 lb. and L is 10 ft.; the maximum bending moment is at the support H and is $50 \times 10 = 500$ lb.-feet. Draw a horizontal line A-A, and from it downward, under the supported end of the beam at H, draw a vertical line b to some convenient scale representing 500 lb.-feet; for instance

100 lb.-feet per inch of length, then line b is 5 in. in length and represents a bending moment of 500 lb.-feet. Now draw a line diagonally upward, joining line A-A at c, which is vertically under W. Supposing it be desired to find the bending moment in the beam at d; draw a perpendicular line from d downward, and the distance e can be scaled off directly and represents the bending moment in the beam at d. If d were 4 ft. from the point of application of the load W and if we scaled length e we would find it to measure 2 in. which represents a bending moment of 200 lb.-feet. We may also calculate it as follows: Length multiplied by load equals $B = 50 \times 4 = 200$ lb.-feet. Since the beam bends concave downward the bending moment is termed negative $(-)$ and the diagram is drawn below line A-A. Of course the actual bending moment or the stress in the metal is the same whether it is drawn positive or negative. Often it is found useful to have a diagram of the shearing stress in the beam. When the imaginary shear at any section takes place so that it moves up on the left and down on the right of said section it is termed positive and when up on the right and down on the left of the section it is negative. In the diagram (Fig. 12), the shear would be + because the portion of the beam at the right of the shear tends to slide down and the part on the left of the shear upward.

The shearing force in this case, neglecting the weight of the beam, is $W = 50$ lb. along the entire length of the beam. Draw a horizontal line C-C and mark off a vertical line equal in length to 50 lb. according to a convenient scale; for instance let 50 lb. = 1 in. Since the shear is the same all along the beam, draw the same vertical line at the other end of the beam and complete the rectangle. In looking at this diagram it may be seen at a glance that near the right end of the beam the bending moment is negligible, therefore, the free end of the beam need only be strong enough to withstand the shearing stress of 50 lb. How to find the actual stress in the metal of the beam, due to the bending moment, will be shown later. (See page 40.) Next consider a cantilever beam, (Fig. 15), loaded at more than one point by loads W_1 and W_2; the maximum bending moment B is at the support H, and it equals the sum of the bending moments due to W_1 and W_2; thus $B = W_1L_1 + W_2L_2$. The bending moment at W_1 is only $W_2\ (L_2 - L_1)$. The hearing stress along the lengths of the beam from W_1 to W_2, $= W_2$, while along the length from H to W_1, it is $W_1 + W_2$, and it is uniform all along said length.

If, for example $W = 100$ lb., $L_1 = 4$ ft., $W_2 = 50$ lb., $L_2 = 6$ ft. Then $B = 100 \times 4 + 50 \times 6 = 700$ lb.-feet. To lay out the bending moment and the shearing force diagram, we may proceed as in the previous case, except that a separate diagram may be drawn (in dotted lines) for the moments due to W_1L_1 and W_2L_2 (the weight of beam is not considered in the example) and afterwards they are added together, *i.e.*, their vertical distances from base line *A-A* are scaled off or measured at a few places, say at *b*, *c*, *d*, and *e*, and are then added; small crosses mark their combined height. The full line is then drawn which represents the combined bending moment. For the shearing force diagram we have $W_1 + W_2$ along the length L_1, and only W_2 between the

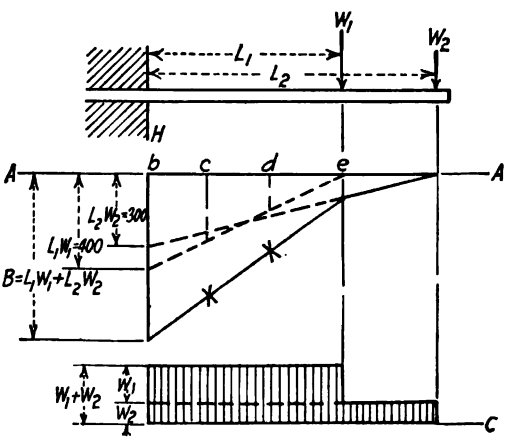


FIG. 15.—Cantilever beam loaded at two points.

points of application of load W_2 and W_1, and the diagram is drawn as shown in (Fig. 15).

If the beam is uniformly loaded, see (Fig. 16), and if the load per unit length is w, the bending moment B would be the same in magnitude as if its entire load were resting at its center of gravity. If the load w is uniform per unit length, the total load is $Lw = W$, and the center of gravity will be at the center of the beam, *i.e.*, at a distance $\frac{1}{2}L$ from the support H. In a cantilever beam the maximum bending moment is always at the support, and it is, $B = \frac{1}{2}L \times Lw = \frac{1}{2}L^2w$, or $\frac{1}{2}LW$, (since $W = Lw$). As an example, suppose the beam to be 8 ft. long and uniformly loaded with 10 lb. per foot of beam; then $W = 8 \times 10 = 80$ lb. and $B = \frac{1}{2} \times 8 \times W = \frac{1}{2} \times 8 \times 80 = 320$ lb.-feet.

The greatest shearing stress is here at H, and is $W = L\,w = 80$ lb. The curve of the bending moment diagram will here not be a straight line, but it will assume the curve of a parabola. At the fixed end of the beam we have seen $B = \frac{1}{2}\,LW = \frac{1}{2}L^2w$. Suppose we wish to find the bending moment at point c, which is l distant from the end. To obtain the bending moment at c consider the entire load between c and d as resting halfway between c and d; the total load here is lw, (if for instance w is 10 lb. per foot and l is 2 ft., the total load is $lw = 2 \times 10 = 20$ lb.), the length is l, and to obtain the bending moment of a uniformly loaded beam we multiply one half the length by the total load,

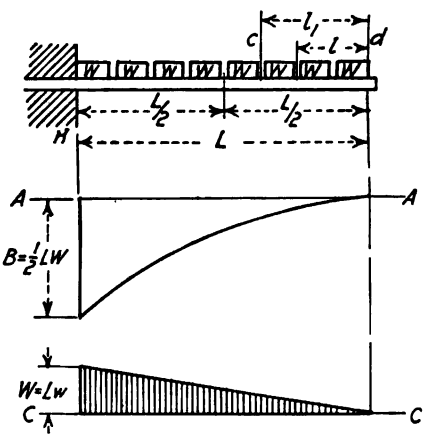


FIG. 16.—Cantilever beam uniformly loaded.

thus $\frac{1}{2}\,l \times lw = \frac{1}{2}l^2w = 20$ lb.-feet. At l_1, B is $\frac{1}{2}\,l^2w$ and by calculating B in this manner at a few points of the beam, we obtain the curve shown. The weight of the beam itself is treated as a uniform load and is added to w per unit length of the beam.

The shearing forces vary from a maximum $Lw = W$ lb. at the support, to zero at free end of the beam.

When there is a concentrated load at one or more points, in addition to a uniform load per unit length, (see Fig. 17) add the maximum bending moment due to the uniform load ($\frac{1}{2}LW$) to the bending moment due to the concentrated load (L_1W_1). Thus the bending moment at H is: $B = \frac{1}{2}LW + L_1W_1$. If the uniform load on the beam equals 15 lb. per foot; if L_1 is 4 ft.; L, 6 ft. and the concentrated load W_1, 150 lb., W is equal to $6 \times 15 = 90$, and the maximum bending moment $B = \frac{1}{2} \times$

$6 \times 90 + 4 \times 150 = 270 + 600 = 870$ lb.-feet. At H the maximum shearing stress $S = W + W_1$. If there are more than one concentrated load in addition to a uniform load they may all be added as was explained under Fig. 15 or as here shown. The curves are first drawn separately for the various loads, and then their vertical distances are added together.

Ordinarily, when calculating the stresses in beams whose length is great, the weight of the beam itself must be considered. In passenger car frames the length between supports, or the end extending beyond the supports, is comparatively small as the

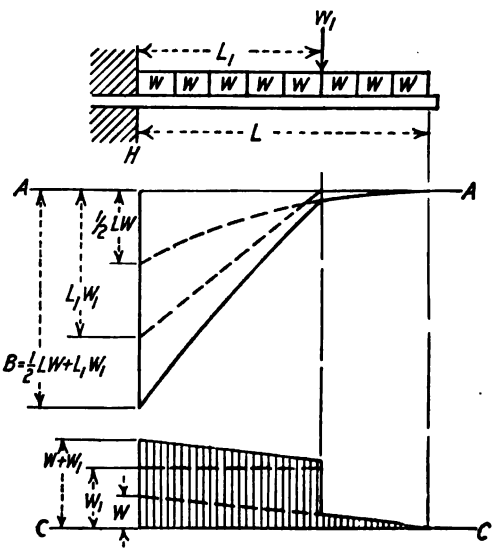


FIG. 17.—Cantilever beam with uniform and concentrated loads.

ratio of load to beam weight is large, hence the weight of the beam itself is often neglected when determining the stresses in the frame.

We will now consider simple beams supported freely at, or near, their ends and loaded between the supports, (see Fig. 18).

The load W pressing the beam down unto the supports is evidently the sum of the reactions; in other words, the sum of the forces at the supports upwards must be equal to the total weight of beam and load. Hence, if W is the downward force or the load, and H_1 and H_2 the reactions at the supports, we may say $W = H_1 + H_2$, and $W - (H_1 + H_2) =$ zero. Thus the

sum of all the forces on the beam is equal to the sum of all the reactions at the supports since the beam is in a state of equilibrium. If the load is concentrated in the center of a beam, supported at two points, the reaction at each support is ½ the total load W. The greatest bending moment B is in the center, and it is equal to the reaction at each support (which is = ½W) multiplied by their distance from i (*i.e.*, ½L), thus $B = ½W \times ½ L = ¼LW$, (when the weight of the beam is neglected). (To make this even more apparent, imagine the beam reversed, *i.e.*, resting on one support in the center, when the length projecting beyond this support is ½L, the load at the end being

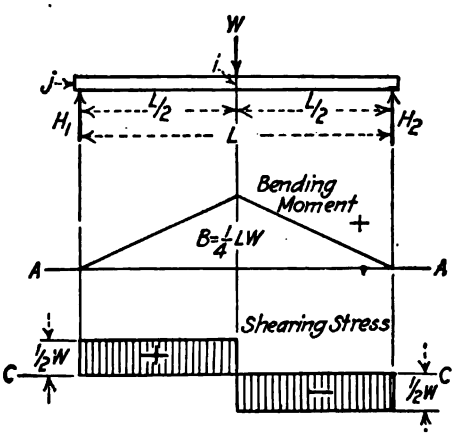


FIG. 18.—Simple beam loaded in center.

½W, which equals H_1 or H_2). The shearing force is uniform over the entire length and = ½W, since ½W is the upward force at each support and this is the force tending to shear the beam in any part from i to the supports.

In drawing the diagrams it must be remembered that the bending moment is a maximum in the center and zero at the ends, and that it varies directly as the distance from the point of application of the load. The load bends the beam so that the concave side is on the top, hence the bending moment is positive. The shearing force is positive between the load W and the left support (see explanation on page 20) hence that part of the diagram is drawn above base line C-C; it is negative to the right of the shear, therefore the right half of the diagram is drawn below

the line. It should be noted that if the two ends of a beam are fixedly attached to some non-yielding structure, as for instance an I-beam attached to two walls of a building, the bending moment will be less, since the ends are restrained in the wall. Most engineering handbooks give the formulas for finding bending moments, etc., of restrained beams. In automobile frames, the members or beams are not considered restrained, as all the frame members are, to a certain extent, yielding. When loads are applied at more than one point or when the load is not in the center, (see Fig. 19) we take moments of the forces, about any convenient point remembering that the algebraic sum of the moments of all the forces is equal to zero.

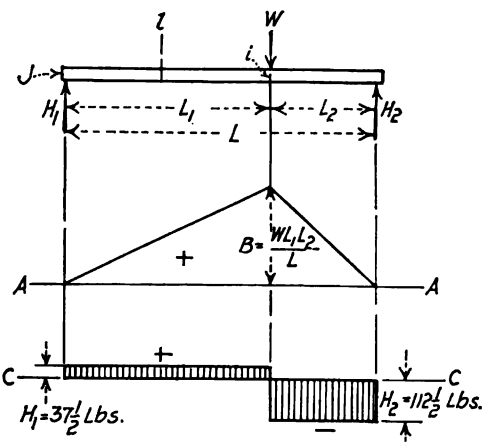


FIG. 19.—Simple beam with load not in center.

In (Fig. 19) first of all, load $W = H_1 + H_2$, *i.e.*, it equals the reactions at the supports, but the reaction of H_1 is not equal to H_2. However, $L_1W - LH_2 = 0$. Action = reaction, and they are always opposed in direction. In this case (assuming for the present that the beam has no weight) L_1W is the moment about point J, in other words, the moment L_1W tends to rotate the beam about point J, while the reaction resisting the rotation about this point J is the force or reaction H_2 (since this is the force which prevents the beam from rotating about J; it holds the beam up) and this force multiplied by L is the moment tending to rotate the beam upward about point J, hence, $L_1W = LH_2$, and $L_1W - LH_2 = 0$; likewise $L_2W - LH_1 = 0$, hence,

$L_1W = LH_2$, from which $H_2 = \frac{L_1W}{L}$; likewise $L_2W = LH_1$, and $H_1 = \frac{L_2W}{L}$. If for example $L_1 = 15$ ft., $L_2 = 5$ ft. (thus $L = 20$ ft.) and $W = 150$ lb; then the reaction at $H_1 = \frac{5 \times 150}{20} = 37\frac{1}{2}$ lb., and $H_2 = \frac{15 \times 150}{20} = 112.5$ lb. The maximum bending moment B is at i and $= \frac{WL_1L_2}{L} = 562.5$ lb.-inches. To make this plain, imagine the beam reversed and resting at one support W, and two loads H_1 and H_2 at the ends. We have found the reaction or the force in lb. at H_1 to be 37.5 lb. and multiplying this by the length L_1 (15) to find the bending moment, we obtain 562.5 lb.-inches. If we take the value found for H_2 and multiply it by the length L_2 we get the same result. The shearing force is uniform from i to the right and equals the reaction $H_2 = 112.5$ lb.; it is also uniform from i to the left and equals reaction $H_1 = 37.5$ lb. Knowing the maximum bending moment at i the bending moment diagram is drawn as shown. It might be stated that the bending moment at any point, for instance l distant from one support, is the reaction from that support multiplied by l. For instance if l is 2 ft. from the left, B is $2 \times 37.5 = 75$ lb.-feet.

The shearing force diagram is as in the last case, positive at the left of the shear, and negative at the right of the shear. The vertical height of the diagram at any point (at any section of the beam) if laid out to some convenient scale as before mentioned will show the force at that point.

Next consider several concentrated loads W_1, W_2, W_3, resting on the beam as shown in Fig. 20. If L_1 is 4 in. L_2, 12 in. L_3, 15 in.; and L, 20 in.; then, taking moments about H_1 we obtain

$$LH_2 = W_1L_1 + W_2L_2 + W_3L_3, \text{ and } H_2 = \frac{W_1L_1 + W_2L_2 + W_3L_3}{L}.$$

Supposing W_1 is 50 lb., W_2, 200 lb. and W_3, 100 lb. we have

$$H_2 = \frac{50 \times 4 + 200 \times 12 + 100 \times 15}{20} = 205 \text{ lb.}$$

In like manner the reaction H_1 (the upward force at the left support) $= \frac{W_3l_3 + W_2l_2 + W_1l_1}{L} = 145$ lb.

The simplest way to find the bending moment at any section of the beam is to construct a bending moment diagram (or else the bending moment may be calculated). To draw the diagram proceed as follows:

Lay out the beam to some scale, for instance, full size, since it is only 20 in. long. Place the loads in their proper positions

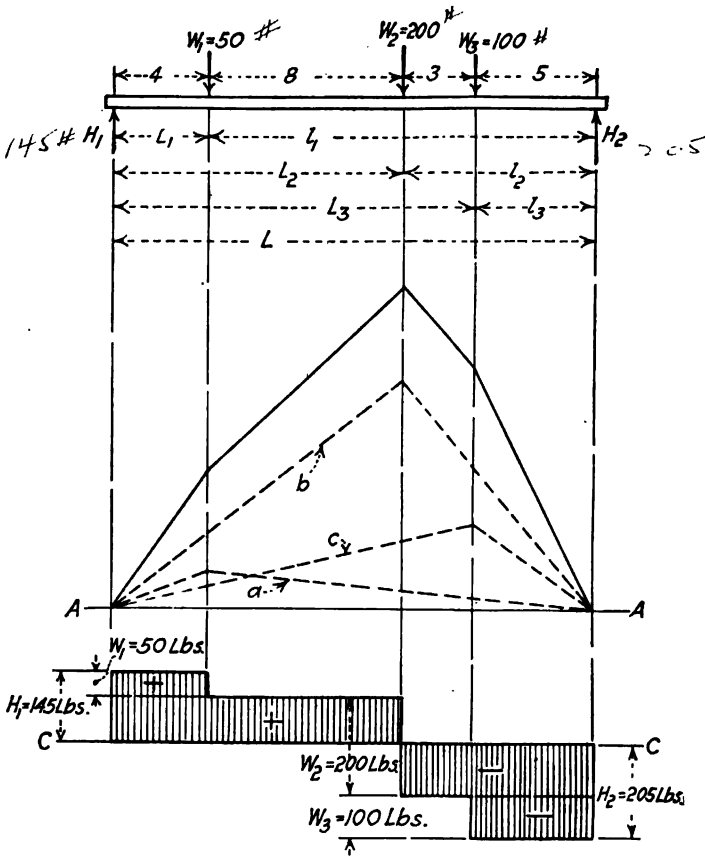


FIG. 20.—Beam with several concentrated loads.

along the beam, to scale, as shown, *i.e.*, $L_1 = 4$; $L_2 = 12$; $L_3 = 15$; $L = 20$. Draw the base line A-A, and draw the bending moment diagram for each load separately as was illustrated in (Fig. 19). In the present case, we have the following bending moments—in the section of the beam under W_1

B (due to load W_1) $= \dfrac{W_1 L_1 l_1}{L} = \dfrac{50 \times 4 \times 16}{20} = 106$ lb.-inches

(Underneath W_2) B(due to W_2) $= \dfrac{W_2 L_2 l_2}{L} =$

$$\frac{200 \times 12 \times 8}{20} = 960 \text{ lb.-inches}$$

(Underneath W_3) B(due to W_3) $= \dfrac{W_3 L_3 l_3}{L} =$

$$\frac{100 \times 15 \times 5}{20} = 375 \text{ lb.-inches}$$

Draw the results to some convenient scale, for instance, let 1 in. be equal to 100 lb.-inches, and complete the triangles, *a*, *b*, *c*, representing the bending moments due to the three loads as shown. Then add the vertical heights of the three diagrams (add their peaks, *i.e.*, the highest points of the triangles, which are underneath the loads $W_1 W_2 W_3$) and draw the diagram shown in full lines. The height of this diagram from the base line can now be measured at any section of the beam, and the answer read off direct; for instance, the highest point will measure 13.4 in. and if the diagram is drawn to a scale of 1 in. equals 100 lb.-inches, the answer will be, maximum bending moments B = 1,340 in.

There is also a simple method for calculating the bending moments, not only under the loads, but at any point or section in a beam.

Bending Moment.—Bending moment at any section as applied to a beam, is the algebraic sum of the individual moments of all the external forces acting on *one* side of the section *only*. Bending moment is said to be positive if the net moment of all the forces on the *left* of the section is clockwise, and negative if the net moment on the *left* of the section is counter-clockwise. The reverse is true in regard to the signs of the bending moment if the forces on the right of the section are considered.

For example, find the bending moment in the section of the beam under W_2 (neglecting the weight of the beam). Beginning *at the left* and taking the moments about point W_2 and marking them positive or negative when they tend to produce clockwise or counter-clockwise rotation respectively, we have the following: (knowing that H_1 = 145 lb. and L_2 = 12).

$$+L_2 H_1 = 145 \times 12 = +1{,}740$$

(This value is positive since it tends to produce clockwise rotation about W_2).

$$-8 \times W_1 = -8 \times 50 = -400$$

(This is negative since its tendency is counter-clockwise).

By finding the algebraic sum of the individual moments, (*i.e.*, adding the positive and deducting the negative values) we obtain the answer, thus $+1{,}740 - 400 = 1{,}340$ lb.-inches.

If we begin from the *right* side when calculating *bending moments*, the positive or negative sign *must be reversed in the answer.*

For instance, starting from the right in the last example (see Fig. 20), we have:—

$$-8H_2 = -8 \times 205 = -1{,}640$$

(H_2 being 205 lb. and l_2 being 8)

$$+3 \times W_3 = 3 \times 100 = 300; \ (W_3 = 100 \text{ lb.})$$

Thus, $-1{,}640 + 300 = -1{,}340$, and *since we started from the right we reverse the minus sign,* hence the result is 1,340 lb.-inches, the same as above.

Suppose now, we wish to find the bending moment in a section of the beam 9 in. from the left support, (which is 5 in. from W_1, and thus between W_1 and W_2). Proceeding as before we obtain

$$+9H_1 = 9 \times 145 = +1{,}305$$
$$-5 \times W^1 = -5 \times 50 = -250$$

and

$$1{,}305 - 250 = 1{,}055 \text{ lb.-inches.}$$

Starting from the right we have (note that $L - 9 = 11$)

$$-11H_2 = -11 \times 205 = -2{,}255$$
$$+6W_3 = +6 \times 100 = +600$$

(9 ft. from H_1 is 6 ft. from W_3)

$$+3W_2 = +3 \times 200 = +600$$

thus $-2{,}255 + 600 + 600 = -1{,}055$, and by reversing the sign, the answer is as before 1,055 lb.-inches. To find the bending moment B in the section of the beam under W_3, we might start from the left, but since we are nearer the right end, the work is shorter when starting from the right, and taking moments about W_3. Thus $B - 5H_2 = -1{,}025$, and by reversing the algebraic sign of the answer we have $B = 1{,}025$ lb.-inches.

Shearing Stress Diagram.—To construct the shearing stress diagram of the beam shown in this figure, draw first base line *C-C* and above it, at the left, draw a vertical line equal to the reaction $H_1 = 145$ lb. (For instance, it may be drawn to a scale of 100 lbs. = 1 in.) At the right of the shear the diagram is negative, hence a vertical line is drawn below base line *C-C*, corresponding to H_2 which is 205 lb. (or 2.05 in. in height). Now draw the shearing stress (the height), corresponding to each load as shown, remembering that all the loads on the top (the weight of the entire beam) must be equal to $H_1 + H_2$. Lay off a height equal to 50 (½ in.) between the left support and W_1; 200 (2 in.) between the line under $W_1$and W_2, and as seen a portion of this height 200 is below the base line, showing it to be negative; in other words, the shearing stress in and near the middle is not 200 but only a portion thereof (since the positive and negative shear counteract each other), it is 145 − 50 = 95 lb. between W_1 and W_2, and $H_2 - 100 = 105$ lb. between W_2 and W_3. At the maximum bending moment (in this case, under W_2), the shearing stress crosses the base line *C-C* and continues uniformly up to a point underneath W_3. From here to the right support the load of W_3 is added *i.e.*, 100 lb. making the shear near the right support 205 lb. Attention is drawn to the fact that always at the maximum bending moment the shear is zero, *i.e.*, in a beam, the maximum bending moment occurs where the shear is situated at, or crosses the zero line. The shearing stress may also be calculated according to the following rule.

Shear.—Shear at any section as applied to a beam, is the algebraic sum of the vertical forces on one side of the section only. If the algebraic sum of the forces on the *left* of the section is upward, the shear is said to be positive. If it is downward on the *left* of the section, the shear is said to be negative. As in the case of the bending moment, the reverse is true when the forces on the right of the section are considered. In both cases of bending moment and of shear, the definition of the positive and negative values is purely conventional, but since these definitions are quite universally used, they should be strictly observed.

It should be remembered that the shear is not the same just to the left or the right of any section (or of a load) while the bending moment *is* the same. Consider for instance, the shear just to the left of the section in the beam under point W_1, or between

the left support and W_1, this equals H_1, which is 145 lb., that is to say, it equals the left reaction, and since it would rise on the left and descend on the right, it is positive. If we started by this method from the right, we have (considering all the positive and negative shears) $-H_2 + W_3 + W_2 + W_1 = -205 + 100 + 200 + 50 = 145$ lb. (When finding the shear the sign is not reversed in the answer whether starting from the right or from the left.)

The shear to the right of W_1 and up to W_2 is $+145 - 50 = 95$ lb.; to check it we may start from the right, $-H_2 + W_3 + W_2 =$

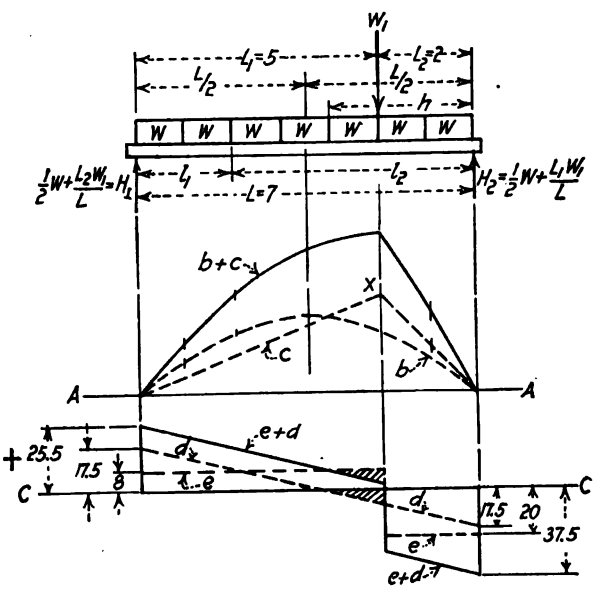


FIG. 21.—Beam with uniform and concentrated loads.

95 lb. Just to the right of W_2 (and up to W_3) we have, starting from the left, $+145 - 50 - 200 = -105$, or starting from the right as a check, $-H_2 + W_3 = -205 + 100 = -105$. Note, at W_2 the shear crosses the zero line, from $+95$ to -105, and as stated before, this occurs where the bending moment is a maximum. To the right of W_3 the shear is simply -205, that is to say in magnitude it equals H_2.

Figure 21 shows a beam uniformly loaded. In this case the greatest bending moment is at the center and is $B = \frac{1}{8} WL$. W being the total weight, ($W = Lw$). As was stated before, the conditions in a beam uniformly loaded (also when consid-

ering the weight of the beam itself) are identical, as far as the maximum bending stresses are concerned, as if ½ the total load or weight W were located at the center of the beam.

The maximum bending moment in the center of the beam shown in (Fig. 18), is ¼ WL, while when uniformly loaded only ½ of W is taken, hence our formula is $B = ½ \times ¼WL = ⅛WL$, which is the maximum bending moment in the middle of the beam, its value being zero at the supports. If a concentrated load W_1 were in *the center* in addition to the uniform load, the moments due to the two loads must be added, and in such case the maximum bending moment $B = ⅛\ WL + ¼\ W_1L$.

If the concentrated load W_1 is not in the center, as shown in (Fig. 21), the curves of the two moments are drawn separately, as shown in this figure.

Consider at first only the uniform load. If this load $w = 5$ lb. per foot and if $L = 7$ ft., then $W = Lw = 35$ lb. $B = ⅛WL = ⅛ \times 35 \times 7 = 30.6$ lb.-feet. in the middle of the beam. The reaction under each support is evidently ½W or 17.5 lb. To find B at any other section for instance, l_1, distant from H_1, we may use the formula $B = w\left(\frac{(l_1 l_2)}{2},\right)$ which is the same as $H_1 \frac{(l_1 l_2)}{L}$. If l_1 is 2 ft., then $B = 5\left(\frac{2 \times 5}{2}\right) = 25$ lb.-feet ($l_2 = 5$). However, a simple method, especially when there are more than one reactions and several loads to consider, is to follow the rule given on page 28. At a distance l_1 for instance, we take the positive bending moment which is $H_1 \times l_1$. The negative bending moment is $l_1\ w \times ½\ l_1$, (the total load to the left of l_1 is $l_1\ w$; and as was stated before, this total load must be considered as located in the center between the end of the beam and the section l_1 distant, thus ½ l_1, and B, due to this uniform load, is therefore $-l_1 w \times ½\ l$). Adding the bending moments algebraically we obtain $+H_1\ l_1 - l_1\ w \times ½\ l_1 = +(17.5 \times 2) - (2 \times 5 \times ½ \times 2) = 35 - 10 = 25$ lb.-feet.

By figuring the bending moments at a few points, the diagram can be drawn and it will be found to follow the curve b (which is that of a parabola).

To draw the shearing stress diagram, curve d, due to the uniform load, we have the maximum value near the supports where it is ½ W, in this case 17.5 lb. In the center of the beam, the shearing stress is zero and crosses the line C-C as shown, since

the shear at the right of (the center) the section would move up. At any other point h distant from the support, the shearing stress S is $H - wh$, or if H tends to produce a negative shear, *i.e.*, if, its tendency is to move up to the right of the section, $S = -H + wh$. This is in accordance with the rule given on page 30. At section h distant, the positive shear, when starting from the right is hw, while the negative shear is $= H_2$. In this case $h = 3$, hence the positive shear is $3 \times 5 = 15$ lb. The negative shear is $17\frac{1}{2}$ lb. and $-17\frac{1}{2} + 15 = 2\frac{1}{2}$ lb. (see dotted line *d*).

The bending moment diagram curve *c*, due to the concentrated load W_1 is similarly derived as that drawn for (Fig. 19). Taking the moment about the right support (see Fig. 21), we have $H_1 L = L_2 W_1$, hence H_1, the reaction at the left support is $\frac{L_2 W_1}{L} =$ 8 lb., and under the right support H_2, it is $\frac{L_1 W_1}{L} = 20$ lb., and this must be added to the reaction due to the uniform load, as shown. Hence, the reaction at the left support is $\frac{1}{2}W + \frac{L_2 W_1}{L}$. If the concentrated load $W_1 = 28$ lb., the maximum bending moment due to this load, as we have seen before, is

$$B = \frac{W_1 L_1 L_2}{L} = \frac{28 \times 5 \times 2}{7} = 40 \text{ lb.-feet.}$$

Mark this value x on the diagram and then draw straight lines to the base line A-A (forming a triangle) as shown, producing curve *c*. By adding the vertical distances of the two curves at a few points along the base line, the resultant bending moment diagram $b + c$ is obtained, and if the curves are drawn to scale, the bending moment at any section of the beam can be found by direct measurement.

The shearing stress due to the concentrated load is, at the right of the section (under W_1), equal to the right reaction = 20 lb., and at the left, 8 lb., (the reaction at H_1). Next draw diagram *e*, which crosses the zero line from -8 to $+20$, underneath W_1. Then add the vertical heights of the two curves and obtain the resultant shown in full lines. It may be noted, when a positive curve is above the base line, and a negative curve directly underneath it, the negative will reduce the height of the positive, and *vice versa* as shown; whichever is the highest

will have a portion remaining. In this case, the shaded portions disappear and it shows that in some portions of a beam, the shear due to a combined load may be smaller than that due to the separate loads. Instead of drawing two separate curves, the resultant bending moment or shearing stress, at any section, may be calculated according to the rules before given. For instance, the bending moment at a section h distant from the right support is found as follows: Starting from the right, the bending moments are $-H_2 \times h + hw \times \frac{1}{2}h + W_1$ times the distance from the section, (in our case this is 1 ft). We have seen that $H_2 = 20 + 17\frac{1}{2} = 37\frac{1}{2}$ lb.; $h = 3$ ft.; $w = 5$ lb; $W = 28$ lb. Hence $B = -37\frac{1}{2} \times 3 + 3 \times 5 \times \frac{1}{2} \times 3 + 28 \times 1 = -105 + 22\frac{1}{2} + 28 =$

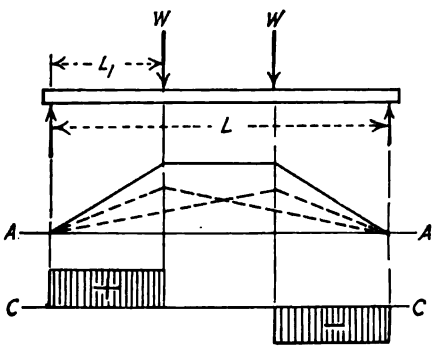


FIG. 22.—Beam with two symmetrical loads.

-62 lb.-feet, and as we started from the right, reverse the sign, hence the answer is $+62$ lb. The next diagram, (Fig. 22) shows two equal loads placed on a beam at equal distance from the supports, the beam being supported at the ends. Here are simply two loads each a certain distance from the end like that shown in (Fig. 19). The problem in this case is very simple, since the loads are symmetrically placed on the beam and therefore, reactions $H_1 + H_2 = 2W$. Hence, either of the reactions equals W and the maximum bending moment is WL_1 and it is constant between the two loads as shown in the diagram. The shearing force is constant between the supports and the points of application of the loads, and equals W. Attention is drawn to the fact that there is no shearing stress in the beam between the two loads, and at the maximum bending moment the shearing force is zero. Figure 23 represents an overhanging beam uniformly loaded; view Q shows the tendency of the beam to bend

under the load. The bending moment and the shearing stress diagrams are also given. The reaction at H_1 can be found by taking moments about H_2, remembering that for this purpose the uniform load can be replaced by a concentrated load in the center of the distance occupied by the uniformly distributed load. The total downward or negative moment about H_2 is the total weight above multiplied by one half the length from H_2, thus $(18w) \times \frac{1}{2}$ 18, and if w is 10 lb. it equals $180 \times 9 = 1{,}620$; and this must be equal to the upward moment $12H_1$. Hence $1{,}620 = 12H_1$, and $H_1 = \frac{1{,}620}{12} = 135$ lb. and $H_2 = 18w -$

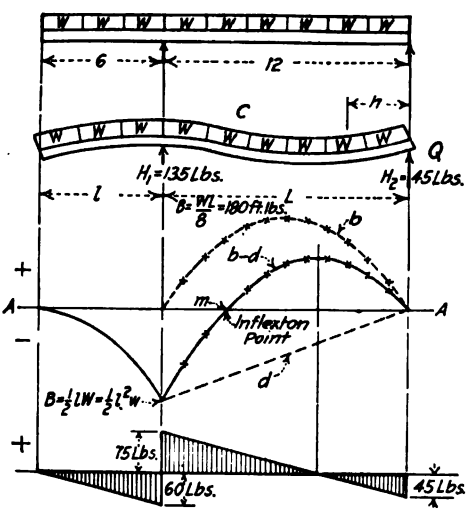


FIG. 23.—Overhanging beam uniformly loaded.

$135 = 180 - 135 = 45$ lb. (Remember that $H_1 + H_2 =$ total load.) The maximum bending moment will here not be in the middle of the beam but its position will depend on the length of the overhang.

The bending moment at any section of the overhang is found in the same manner as that of a uniformly loaded cantilever beam (see Fig. 16). At any other section (between the two supports) it may be calculated as shown under the next figure. The shearing force at H_2 equals the reaction = 45 lb. and is negative, thus −45 lb. At any other point between the reactions, h distant from a reaction H, it equals $H - wh$, or if the

reaction moves up at the right of the section as is here the case, it is $-H_2 + wh$. At H_1, the positive shear is therefore $-45 + 10 \times 12 = 75$ lb. The maximum negative shear, due to the overhanging load, is $lw = 6 \times 10 = 60$ lb.

From the curves it is seen that at the maximum bending moment of the beam, between the supports, the shear is zero, and that it again crosses the zero line at H_1, where there is another maximum bending moment. It is important to note that between the supports the bending moment crosses from a positive to a negative value, due to the overhang. Evidently the stress in the outer fibers of the beam changes from tension to compression and *vice versa*, and this change takes place at the zero line, marked *m*, where the bending moment is zero. The point where the stress changes from tension to compression, or *vice versa*, is called the inflexion point.

Instead of figuring the bending moment for the various portions of the beam, the resultant may be derived graphically, by first plotting curve *b* (shown dotted), the same as curve *b* was plotted in (Fig. 21) that is to say, plot the curve due to the uniform load (disregarding for the present the overhang). Then draw a straight line *d* (shown dotted) from the value of bending moment underneath H_1, to the bending moment underneath H_2 (which here is zero as there is no overhang to the right of H_2). By adding *b* and *d* algebraically we obtain the resultant curve. Since *d* is below the line, it is negative, hence the resultant is $b - d$, *viz.* the height of *d* from the base line is deducted from the height of curve *b*. Evidently at the spot where *b* and *d* are the same distance from the base line, one above, the other below, the resultant will cross the zero line, which here is at point *m*, and is called the inflexion point.

If there were another overhang beyond the right support, (see Fig. 24), first plot the bending moment, curve *b*, due to the load between the supports, then plot the bending moment due to each overhang *c* and *e*. Next draw a straight line *d* from the bending moments under H_1 to that of H_2, and by deducting curve *d* from curve *b*, the resultant $b - d$ is obtained as in the last case. The shearing stress diagram can be obtained graphically as shown. The shearing stress due to the left overhang is the total weight on the overhang $wL_1 = 60$ lb. and is negative just to the left of H_1. The positive shear just to the right of H_1, is $H_1 - 60$.

In order to find the reactions at H_1 and H_2 proceed as follows:

First find the moments about one of the ends of the beam, say about O.

It was stated previously that a uniform load over a given length may be represented for this purpose by a concentrated load in the center of such length or distance. Take for instance the load on the left overhang $wL_1 = W_1 = 60$ lb., (w being 10 lb.

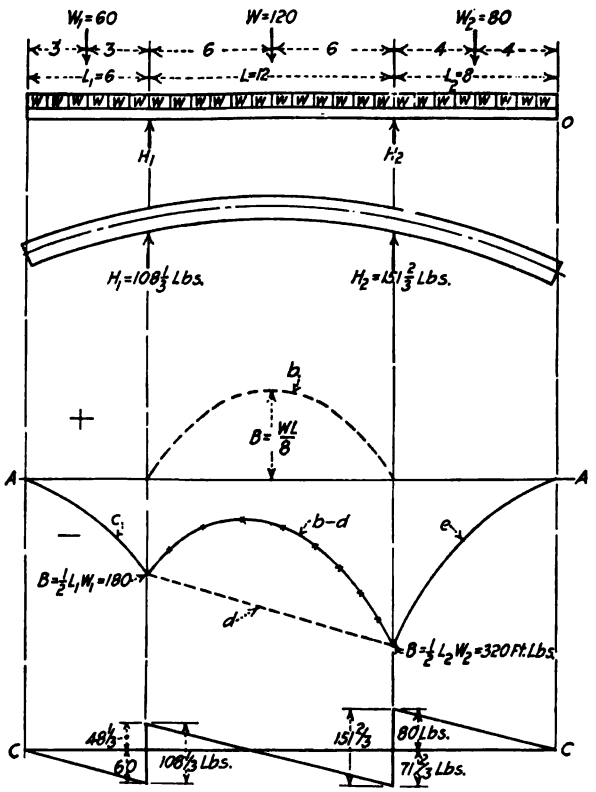


FIG. 24.—Beam uniformly loaded; two ends overhanging.

per foot). This uniform load wL_1 will produce the same moment about H_1 (or about O) as the entire load of 60 lb., concentrated in the middle of the overhang. The same thing holds good for the center span where the uniform load may be represented by W, and on the right overhang by W_2. Hence, the moments which tend to produce clockwise rotation about O are: H_1 times length to O, $+H_2$ times length to O, and this equals the moments tending to produce counter-clockwise rotation, *viz.*, W_1 times

length to O, $+W$ times length to O, $+W_2$ times length to O. (The uniform load is in this example 10 lb. per foot. Therefore $W_1 = 6 \times 10 = 60$ lb.; $W = 12 \times 10 = 120$ lb. and $W_2 = 80$ lb.)

Thus $20H_1 + 8H_2 = 23 \times 60 + 14 \times 120 + 4 \times 80 = 1,380 + 1,680 + 320 = 3,380$. But the total load on the beam is $W_1 + W + W_2 = 60 + 120 + 80 = 260$ lb. and $H_1 + H_2$ must be equal to 260 lb. We here have a simple simultaneous equation which can be solved by algebra as follows:

$$H_1 + H_2 = 260$$

Hence

$$H_2 = 260 - H_1$$

Substituting this value for H_2 in the equation $20H_1 + 8H_2 = 3,380$, we write:

$$\begin{aligned} 20H_1 + 8\,(260 - H_1) &= \\ 20H_1 + 2,080 - 8H_1 &= \\ 12H_1 + 2,080 &= 3,380, \end{aligned}$$

then

$$H_1 = \frac{3,380 - 2,080}{12} = \frac{1,300}{12} = 108\tfrac{1}{3} \text{ lb.}$$

In a similar manner H_2 can be found (taking $H_1 = 260 - H_2$) to equal $151\frac{2}{3}$ lb., or we may write $H_2 = 260 - H_1 = 260 - 108\frac{1}{3} = 151\frac{2}{3}$ lb.

Where there are only two supports as in (Fig. 24), the reactions may also be found without a simultaneous equation.

For instance, we may start by taking moments about H_1, starting from the left and taking the moments which tend to produce a clockwise rotation as positive, and those counter-clockwise, as negative.

First establish the center of gravity of the uniform load as seen before, in the center of the distances, L_1, L, and L_2, and mark their distances from the supports, for instance W_1 is 3 ft. from H_1, and so on for the rest of the beam. Then taking moments about H_1 we write:

$$-3 \times 60 = -180$$

(This quantity is "minus" since it tends to produce counter-clockwise rotation).

$$\begin{aligned} +6 \times 120 &= 720 \\ -12 \times H_2 &= -12H_2 \\ +16 \times 80 &= 1,280, \end{aligned}$$

(W_2 being 16 ft. from H_1.) By adding all the positive and negative values algebraically we have,

$$2{,}000 - 180 - 12H_2 =$$

or

$$+1{,}820 - 12H_2 = O$$

hence

$$1{,}820 = 12H_2$$

and

$$H_2 = \frac{1{,}820}{12} = 151\tfrac{2}{3} \text{ lb.}$$

In this manner the moments about H_2 would give for the reaction at H_1, 108 ⅓ lb.

A similar method may be pursued for finding the bending moment B for any section of a beam.

Suppose we wish to find the bending moment at H_1, (Fig. 24).

Starting from the left we have $-\frac{1}{2} L_1 \times$ total weight of the left overhang $= -3 \times 60 = -180$ lb.-feet. Let us see how this checks by starting from the right, taking, of course, the center of gravity of the uniform load at ½ the distance from H_1, but the reaction H_2 is multiplied by the total length from H_1 since the reaction may be compared to a concentrated load. Thus the total length from the right end of the beam to H_1 is 20 ft. and the total load on this 20 ft. is $20 + 10 = 200$ lb. The center of gravity of this load is 10 ft. from H_1, therefore, we have $+10 \times 200 = 2{,}000$ lb.-feet. The reaction H_2, (which is the same as a load or a force tending to produce counter-clockwise rotation about H_1) is 12 ft. away, thus $-12 \times 151\frac{2}{3} = -1{,}820$; deducting this from the positive moment, we have $2{,}000 - 1{,}820 = 180$, and as mentioned before, when starting from the right the sign is reversed, hence the answer is at H_1, $B = -180$ lb.-feet.

Finding the bending moment at a point say 5 ft. from the left reaction which is 11 ft. from the left end of the beam, (the total load on this 11 ft. is 11×10 lb. $= 110$ lb.) we write: B, (due to uniform load above) $= -\frac{1}{2} \times 11 \times 110$ lb. $= -605$; and due to the upward pressure from the reaction, $B = +5 \times H_1 = +5 \times 108\frac{1}{3} = +541\frac{2}{3}$; (deduct this from -605).
Answer, $-63\frac{1}{3}$ lb.-feet.

Finding B from the right, at the same section (note that total length of beam at the right of this point is $26 - 11 = 15$ ft.), we have $\frac{1}{2} \times 15 \times 15 \times 10 = +1{,}125$ lb.-feet and the

negative bending moment is $-7 \times H_2 = 7 \times 151\frac{2}{3} = -1{,}061\frac{2}{3}$; deducting this from $+1{,}125$, and reversing the sign, we obtain the answer $-63\frac{1}{3}$ lb.-feet as before. In the uniformly loaded beam shown in (Fig. 24), if the center span were lengthened the bend in the beam (the negative bending moment) would first decrease until it reached the base line; if it were further lengthened the bending moment would rise above the base line and thus become positive, and the bend in the beam between the supports would appear as in the last figure.

The shear in (Fig. 24) will vary constantly since we are dealing with a distributed load instead of a concentrated load. For

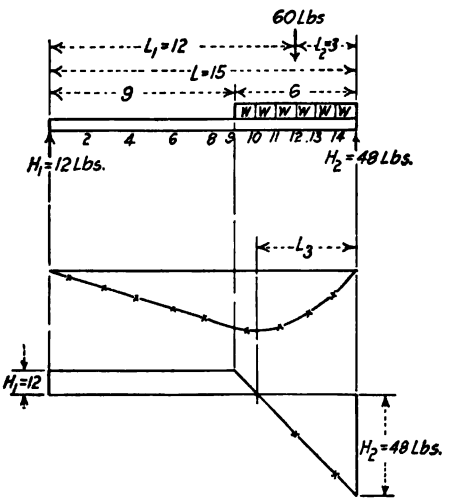


FIG. 25.

instance, just to the left of H_1, the shear is -60 lb., while to the right of H_1 it is $-60 + 108\frac{1}{3} = 48\frac{1}{3}$ lb., that is to say, the load above (60 lb.) would tend to produce a negative shear and is therefore minus, while H_2, ($108\frac{1}{3}$ lb.) would tend to produce a positive shear. Next, find the shear at a point 5 ft. to the right of H_1. The total load to the left of this point $11 \times 10 = 110$ lb. The shear is $-110 + 108\frac{1}{3} = -1\frac{2}{3}$ lb. If taken from the right we have (the total load to the right of this point is $15 \times 10 = 150$ lb.) $150 - 151\frac{2}{3} = -1\frac{2}{3}$, showing that it checks. It is only under a concentrated load or over a support or reaction that the shear will show a great sudden change.

Figure 25 shows a simple beam uniformly loaded for a portion

of its length, near the right supports. The total length is 15 in. the uniform load is spread over 6 in., and is 10 lb. per inch. To find the reactions at the supports first find the center of gravity of the uniformly distributed load, and this is 60 lb. located 3 in. from the right support as shown. By the principle of moments, we have $H_1 \times 15 = 3 \times 60$, or $H_1 = \frac{3 \times 60}{15} = 12$ lb.; $H_2 = 60 - 12 = 48$ lb. (or $H_2 = \frac{L_1 \times 60}{L} = \frac{12 \times 60}{15} = 48$ lb.)

(If the weight of the beam were considered, and let us say it weighed 1 lb. per inch of length, or a total of 15 lb., then $H_1 = 12 + \frac{1}{2}$ of $15 = 19.5$ lb., and $H_2 = 55.5$ lb.)

Suppose it is desired to find the bending moment at a number of points along the beam; starting from the left, and neglecting the weight of the beam, a simple method of procedure is as follows:

2 in. from the left support, $B = 2 \times 12 = 24$ lb.-inches (H_1 being 12 lb.)
4 in. from the left support, $B = 4 \times 12 = 48$ lb.-inches
6 in. from the left support, $B = 6 \times 12 = 72$ lb.-inches
8 in. from the left support, $B = 8 \times 12 = 96$ lb.-inches
9 in. from the left support, $B = 9 \times 12 = 108$ lb.-inches

So far we had no load or force above it to consider, but at a section 10 in. from the left we have a load of 10 lb., whose center of gravity is $\frac{1}{2}$ in. from the section and which tends to produce counter-clockwise rotation, hence $B = 10 \times 12 - \frac{1}{2} \times 10 = 115$ lb.-inches.

Eleven inches from the left, $B = 11 \times 12 - 1 \times 20 = 112$ lb.-inches, that is to say, there now is a load of 10 lb. per inch or 20 lb. acting downward, whose center of gravity is $\frac{1}{2}$ of 2 in., or 1 in. away from the section where the bending moment is to be found. The next point is 12 in. from the left, and the load acting downward is 30 lb. and its center of gravity is $1\frac{1}{2}$ in. from said point, and its counter-clockwise moment is $-1\frac{1}{2} \times 30$; thus at

12 in. from the left, $B = 12 \times 12 - 1\frac{1}{2} \times 30 = 99$ lb.-inches
13 in. from the left, $B = 13 \times 12 - 2 \times 40 = 76$ lb.-inches
14 in. from the left, $B = 14 \times 12 - 2\frac{1}{2} \times 50 = 43$ lb.-inches
15 in. from the left, $B = 15 \times 12 - 3 \times 60 = 0$

Plotting a curve with the values so found gives the bending moment diagram. As seen the maximum bending moment is here 10 in. from the left support, or 5 in. from the right end, and

amounts to 115 lb.-inches. This may be calculated more accurately by the formula

$$B_{max} = \frac{2wL_1^2L_2^2}{L^2} = \frac{2 \times 10 \times 12 \times 12 \times 3 \times 3}{15 \times 15} = 115.2 \text{ lb.-inches,}$$

and to find the distance L_3 from the right at which the maximum bending moment occurs we may use the formula:

$$L_3 = \frac{2L_2L_1}{L} = \frac{72}{15} = 4.8 \text{ in.}$$

It was shown before that at the maximum bending moment, the shearing stress crosses the zero line. The shearing stress between H_1 and the beginning of the uniformly distributed load is uniform and equal to $H_1 = 12$ lb. (If the weight of the beam is to be considered, the shear due to its weight must be added as was shown in Fig. 21.) Since the force or the reaction H_1 tends to move upwards it produces a positive shear. Beginning 9 in. from the left, where the uniform load starts, the latter must be added:

At 10 in. from the left, shear = +12 −10 = 2 lb.
At 11 in. from the left, shear = +12 −20 = − 8 lb.
(or starting from the right we have −48 + (4 × 10) = −8)
At 12 in. from the left, shear = +12 −30 = −18 lb.
At 14 in. from the left, shear = +12 −50 = −38 lb.
At 15 in. from the left, shear = +12 −60 = −48 lb.
that is to say, it equals the right reaction H_2.

The shear diagram is then drawn with these values as shown. See Chapt. III for figuring stresses in complete chassis frame side members.

Moment of Resistance.—The tension or compression stresses in the fibers of a beam are proportional to their distance from the neutral surface, as was shown on page 18. The greater this distance, the greater is the strength of the beam. The stresses produced in the structure of the beam when loaded, produce a counter-moment in the beam, which resists the bending action or the bending moment, and this is called the moment of resistance of the beam. Since action equals reaction, the bending moment equals the moment of resistance.

Moment of Inertia.—The moment of inertia of a cross-sectional area, or of a surface, is a measure or function of the moment of resistance and therefore it is indicative of the strength of the beam. The moment of inertia equals the products obtained

by multiplying each elementary area by the square of its distance from the gravity axis. Suppose a_1, (Fig. 26), is a small particle of an area at a distance r_1 from the neutral surface; the moment of inertia I of the small area $a_1 = a_1 r_1^2$. The moment of inertia of the area $a_2 = a_2 r_2^2$. The moment of inertia of the entire section equals the sum A of all the elementary areas (each considered minute) multiplied by the square of the mean distance r of all the small areas from the neutral axis, thus $I = Ar^2$. To compute the moment of inertia of a section requires the use of calculus, which is beyond the scope of this work. Moments of inertia of various sections are found on pages 46 to 49.

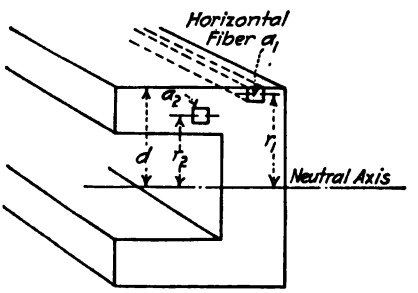


FIG. 26.—Illustrating moment of inertia.

If a beam is loaded or stressed and if d is the distance of the outermost fiber which resists the stress, from the neutral surface, and if S is the unit stress (stress per square inch) in the outermost fiber, having an area a, the stress in each of the outermost fibers is, $a\,S$.

If we take an area a_2 nearer the neutral axis, say at distance r_2, the stress in the area, (as compared to the stress in the outermost area d distant) is $a_2 S \frac{r^2}{d}$. The moment of a force is the magnitude of the force multiplied by its distance from the axis; in our case the distance is r_2, since we are considering the stress or the force acting on area a_2. This stress or force is zero at the neutral surface; on the area a_2 it is $a_2 S \frac{r_2}{d}$, and the moment of the force on a_2, is $a_2 S \frac{r_2}{d} r_2 = \frac{a_2 S r_2^2}{d}$; which may also be written $\frac{S}{d} a_2 r_2^2$. If, as before, A stands for the sum of all the small areas (into which the entire section is divided) and r^2 for the sum of the square of the mean distance of these areas, then the moment of the stress in the entire section induced by a load on the beam $= \frac{S}{d} Ar^2$, and this is called the moment of resistance, and it is of course equal to the bending moment B. It was shown before

that $Ar^2 = I$, the moment of inertia of the section, hence we may also write the moment of resistance of a section $= \frac{SI}{d} = S\frac{I}{d}$, and this equals the bending moment B for that section. The value $\frac{I}{d}$ is often used in engineering handbooks and is called the section modulus, as a rule designated by z, hence bending moment $B = S\frac{I}{d} = Sz$, from which $S = \frac{B}{Z}$.

These formulas are found very useful in engineering calculations, for finding the necessary section to carry a given load, finding the fiber stress in the metal, etc. remembering that S = the stress (in pounds per square inch) in the fibers on the upper or the lower side of the beam. The fiber stress half way between the neutral axis and the outermost fibers, *i.e.* at distance $\frac{d}{2}$ is equal to $\frac{S}{2}$, in other words, the stress is proportional to the distance from the neutral axis. The moment of inertia I of a section is dependent on the shape of the section (how the small areas are placed or where they are located) and on the square of the average distance r of all the small areas from the neutral axis. If we for instance, take a square bar of a cross-section of one square inch, the mean distance r of the small areas would be less than in a channel section having a cross-sectional area of one square inch, as the height of the channel is considerably more than one inch. Hence, an ordinary channel will carry a much greater load than a solid beam of an equal cross sectional area. The neutral surface is always passing through the center of gravity of the section, and is always considered as running parallel to the top and bottom lines of the beam.

The moment of inertia is expressed in so many inch^4, as both, the area is expressed in square inches or inch^2, and "r" is squared ($I = Ar^2$) hence I is expressed in inch^4.

Examples.—Find the maximum load a beam 5 ft. long will safely carry on one end, when the other end is fixed (a cantilever beam). The factor of safety may be taken as 5, and the ultimate strength of the material of the beam, let us say, as 80,000 lb. per square inch; therefore, stress S, to safely carry the load is $\frac{80{,}000}{5} = 16{,}000$ lb. per square inch. If the width of the beam is

2 in., the height of the metal 5 in., $L = 5$ ft. or 60 in., the permissible fiber stress $S = 16{,}000$ lb., the bending moment $B = \frac{SI}{d} = LW$, for bending moment equals length multiplied by load. From the table on page 46 it can be seen that the moment of inertia I of a rectangle is $\frac{bh^3}{12} = \frac{2 \times 5 \times 5 \times 5}{12} = 20.8$, where b is the width, and h the height of the beam or the rectangle; d equals one-half of the height, *i.e.*, 2.5 in., the length L is 60 in. The problem is to find W.

It was shown before that the bending moment equals $\frac{SI}{d} = LW$; substituting the figures for the symbols we have

$$\frac{16{,}000 \times 20.8}{2.5} = 60W,$$

from which

$$W = \frac{16{,}000 \times 20.8}{2.5 \times 60} = 2{,}218.7 \text{ lb.}$$

If W is known and we wish to find S, from the foregoing equation we can write

$$S = \frac{LWd}{I} = \frac{60 \times 2{,}218.7 \times 2.5}{20.8} = 16{,}000 \text{ (lb. per square inch).}$$

The section modulus $Z = \frac{I}{d}$, and d is $\frac{h}{2}$. For a rectangle, $I = \frac{bh^3}{12}$, and $Z = \frac{I}{d} = \frac{(bh^3/12)}{d} = \frac{bh^2}{6}$ = section modulus of a rectangle. As a rule the work is shortened by making use of the section modulus; most engineering handbooks give its numerical value for the different cross-sectional areas. On pages 46 to 49 are tables[1] containing such values.

Summarizing the foregoing, we may state: If B is the bending moment; I the moment of inertia of the cross-section; d the distance from the neutral surface to the extremity of the section (in the tables on pages 46 to 49, X and X_1 are used); Z the section modulus; S the fiber stress per unit area or the permissible working stress in pounds per square inch, then, from the fundamental equation for beams we have:—$B = \frac{SI}{d}$, $S = \frac{Bd}{I}$ and

[1] From the handbook of the Cambria Steel Co., prepared by George F. Thackeray.

since $\frac{I}{d} = Z$, $B = SZ$, and the bending moment divided by the fiber stress or the permissible working stress equals the section modulus, thus $Z = \frac{B}{S}$; $S = \frac{B}{Z}$.

TABLE III

Section.	Area of Section. A	Distance from Neutral Axis to Extreme Fiber of Section. z and z_1	Moment of Inertia. I	Section Modulus. $Z = \frac{I}{z}$	Radius of Gyration. $r = \sqrt{\frac{I}{A}}$
	a^2	$z = z_1 = \frac{a}{2}$	$\frac{a^4}{12}$	$\frac{a^3}{6}$	$\frac{a}{\sqrt{12}} = .288a$
	a^2	$z = z_1 = a$	$\frac{a^4}{3}$	$\frac{a^3}{3}$	$\frac{a}{\sqrt{3}} = .577a$
	$a^2 - a_1^2$	$z = z_1 = \frac{a}{2}$	$\frac{a^4 - a_1^4}{12}$	$\frac{a^4 - a_1^4}{6a}$	$\sqrt{\frac{a^2 + a_1^2}{12}}$
	a^2	$z = z_1 = \frac{a}{\sqrt{2}} = .707a$	$\frac{a^4}{12}$	$\frac{a^3}{6\sqrt{2}} = .118a^3$	$\frac{a}{\sqrt{12}} = .288a$
	bd	$z = z_1 = \frac{d}{2}$	$\frac{bd^3}{12}$	$\frac{bd^2}{6}$	$\frac{d}{\sqrt{12}} = .2864$
	bd	$z = z_1 = d$	$\frac{bd^3}{3}$	$\frac{bd^2}{3}$	$\frac{d}{\sqrt{3}} = .577d$
	$bd - b_1d_1$	$z = z_1 = \frac{d}{2}$	$\frac{bd^3 - b_1d_1^3}{12}$	$\frac{bd^3 - b_1d_1^3}{6d}$	$\sqrt{\frac{bd^3 - b_1d_1^3}{12(bd - b_1d_1)}}$
	bd	$z = \frac{bd}{\sqrt{b^2+d^2}}$	$\frac{b^3d^3}{6(b^2+d^2)}$	$\frac{b^2d^2}{6\sqrt{b^2+d^2}}$	$\frac{bd}{\sqrt{6(b^2+d^2)}}$

(From Cambria Steel Handbook prepared by George F. Thackeray.)

Example.—In a simple cantilever beam, see (Fig. 11), the distance L is 48 in.; the load W 1,000 lb.; the max. bending moment is $1{,}000 \times 48 = 48{,}000$ lb.-inches. If the ultimate strength of the material is 80,000 lb. per square inch (under tension and compression) and if we assume a factor of safety of 5, the permissible

working stress in the metal will be $\frac{80,000}{5} = 16,000$ lb. per square inch. We thus have $B = 48,000$; $S = 16,000$, therefore $Z = \frac{48,000}{16,000} = 3$. Table II gives the section modulus of an

TABLE III.—*Continued*

Section.	Area of Section. A	Distance from Neutral Axis to Extremities of Section. z and z_1	Moment of Inertia. I	Section Modulus. $Z = \frac{I}{z}$	Radius of Gyration. $r = \sqrt{\frac{I}{A}}$
	bd	$z = \frac{d\cos\alpha + b\sin\alpha}{2}$	$\frac{bd}{12}(d^2\cos^2\alpha + b^2\sin^2\alpha)$	$\frac{db}{6}\left(\frac{d\cos\alpha + b\sin\alpha}{d\cos\alpha + b\sin\alpha}\right)$	$\sqrt{\frac{d^2\cos^2\alpha + b^2\sin^2\alpha}{12}}$
	$\frac{bd}{2}$	$z = \frac{d}{3}$ $z_1 = \frac{2d}{3}$	$\frac{bd^3}{36}$	$\frac{bd^2}{24}$	$\frac{d}{\sqrt{18}} = .2357d$
	$\frac{bd}{2}$	$z = d$	Axis through base: $\frac{bd^3}{12}$ Axis through apex: $\frac{bd^3}{4}$	$\frac{bd^2}{12}$ $\frac{bd^2}{4}$	$\frac{d}{\sqrt{6}} = .4082d$ $\frac{d}{\sqrt{2}} = .7071d$
	$.785d^2$	$z = \frac{d}{2}$	$.049d^4$	$.098d^3$	$\frac{d}{4}$
	$.785(d^2 - d_1^2)$	$z = \frac{d}{2}$	$.049(d^4 - d_1^4)$	$.098\frac{(d^4 - d_1^4)}{d}$	$\frac{\sqrt{d^2 + d_1^2}}{4}$
	$.393d^2$	$z = \frac{2d}{3\pi} = .5234d$ $z_1 = \frac{(3\pi - 4)d}{6\pi} = .2234d$	$.0068d^4$	$.0259d^3$	$\frac{\sqrt{9\pi^2 - 64}}{12\pi} \cdot d = .1305d$
	$\frac{b+b_1}{2} \cdot d$	$z = \frac{b+2b_1}{b+b_1} \cdot \frac{d}{3}$ $z_1 = \frac{b_1+2b}{b+b_1} \cdot \frac{d}{3}$	$\frac{b^2+4bb_1+b_1^2}{36(b+b_1)} \cdot d^3$	$\frac{b^2+4bb_1+b_1^2}{12(b_1+2b)} \cdot d^2$	$\frac{d}{6(b+b_1)}\sqrt{2(b^2+4bb_1+b_1^2)}$

ordinary rectangular section as $\frac{bh^2}{6}$, where b is the width and h[1] the height. In our example we may therefore write, $Z = 3 = \frac{bh^2}{6}$, and if we assume the beam width to be 1 in., from the last

[1] In the table d is used to denote the height.

equation we find $h^2 = \frac{3 \times 6}{b} = \frac{18}{1}$, and $h = \sqrt{18} = 4.25$. Therefore, to support such a load properly on a cantilever beam 48 in. long with a factor of safety of 5, that is to say, with a working

TABLE III.—*Continued*

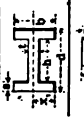
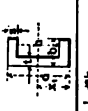
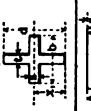
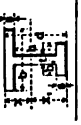
Sections.	Area of Section. A	Distance from Neutral Axis to Extremities of Section. z and z₁	Moment of Inertia. I	Section Modulus. $Z = \frac{I}{z_1}$	Radius of Gyration. $r = \sqrt{\frac{I}{A}}$
	$\frac{3}{2} d^2 \tan 30° = .866 d^2$	$z_1 = \frac{d}{2}$	$\frac{A}{12}\left[\frac{d^2(1+2\cos^2 30°)}{4\cos^2 30°}\right] = .068 d^2$	$\frac{A}{6}\left[\frac{d(1+2\cos^2 30°)}{4\cos^2 30°}\right] = .125 d^3$	$\frac{d}{4\cos 30°}\sqrt{\frac{1+2\cos^2 30°}{3}} = .264 d$
	$\frac{3}{2} d^2 \tan 30° = .866 d^2$	$z_1 = \frac{d}{2\cos 30°} = .577 d$	$\frac{A}{12}\left[\frac{d^2(1+2\cos^2 30°)}{4\cos^2 30°}\right] = .068 d^4$	$\frac{A}{6}\left[\frac{d(1+2\cos^2 30°)}{4\cos 30°}\right] = .104 d^3$	$\frac{d}{4\cos 30°}\sqrt{\frac{1+2\cos^2 30°}{3}} = .264 d$
	$2 d^2 \tan 22\frac{1}{2}° = .828 d^2$	$z_1 = \frac{d}{2}$	$\frac{A}{12}\left[\frac{d^2(1+2\cos^2 22\frac{1}{2}°)}{4\cos^2 22\frac{1}{2}°}\right] = .058 d^4$	$\frac{A}{6}\left[\frac{d(1+2\cos^2 22\frac{1}{2}°)}{4\cos 22\frac{1}{2}°}\right] = .109 d^3$	$\frac{d}{4\cos 22\frac{1}{2}°}\sqrt{\frac{1+2\cos^2 22\frac{1}{2}°}{3}} = .357 d$
	$\frac{\pi bd}{4} = .785\ bd$	$z_1 = \frac{d}{2}$	$\frac{\pi bd^3}{64} = .049 bd^3$	$\frac{\pi bd^2}{32} = .098 bd^2$	$\frac{d}{4}$
	$bd + 2b'(s + s')$	$z_1 = \frac{d}{2}$	$\frac{1}{12}\left[bd^3 - \frac{1}{4g^2}(h^4 - 1^4)\right]$ where $g = \frac{h-1}{b-1}$	$\frac{2I}{d}$	$r = \sqrt{\frac{I}{A}}$
	$bd + 2b'(s + s')$	$z_1 = \frac{b}{2}$	$\frac{1}{12}\left[b^3(d-h)+h^3 + \frac{1}{4g^2}(h^4 - 1^4)\right]$ where $g = \frac{h-1}{b-1}$	$\frac{2I}{b}$	$r = \sqrt{\frac{I}{A}}$
	$bd + b'(s + s')$	$z_1 = \frac{d}{2}$	$\frac{1}{12}\left[bd^3 - \frac{1}{6g^2}(h^4 - 1^4)\right]$ where $g = \frac{h-1}{2(b-1)}$	$\frac{2I}{d}$	$r = \sqrt{\frac{I}{A}}$
	$bd + b'(s + s')$	$x = \frac{bd^2 + hh'^2 + \frac{s}{3}(b-b')^2}{2A}$, $z_1 = b - x$	$\frac{1}{3}\left[2bx^3 + h'^3 + \frac{s}{2}(b^4 - b'^4)\right] - Ax^2$ where $s = \frac{h-1}{2(b-1)}$	$\frac{I}{b-x}$	$r = \sqrt{\frac{I}{A}}$

stress of 16,000 lb. per square inch, the height of the beam must be 4.25 in.

From the formula for the section modulus it can also be seen that the capacity of a beam to resist bending is proportional to the square of its height h. Hence, by doubling the height of a beam,

we increase its moment of resistance (its ultimate strength in the direction of the height or depth) to four times its original strength. By doubling the width b, we only double its strength. Increasing the height h however, also increases its tendency to twist or buckle, and this must be considered when designing beams or

TABLE III.—*Continued*

Sections.	Area of Section. A	Distances from Neutral Axis to Extreme Fibres of Section. x and n	Moment of Inertia. I	Section Modulus. $Z = \frac{I}{n}$	Radius of Gyration. $r = \sqrt{\frac{I}{A}}$
	$bd - b_1(b - t)$	$n = \frac{d}{2}$	$\frac{bd^3 - b_1(b - t)}{12}$	$\frac{bd^3 - b_1(b - t)}{6d}$	$\sqrt{\frac{bd^3 - b_1(b - t)}{12[bd - b_1(b - t)]}}$
	$bd - b_1(b - t)$	$n = \frac{b}{2}$	$\frac{2hb^3 + h_1b^3}{12}$	$\frac{2hb^3 + h_1b^3}{6b}$	$\sqrt{\frac{2hb^3 + h_1b^3}{12[bd - b_1(b - t)]}}$
	$bd - b_1(b - t)$	$n = \frac{d}{2}$	$\frac{bd^3 - b_1(b - t)}{12}$	$\frac{bd^3 - b_1(b - t)}{6d}$	$\sqrt{\frac{bd^3 - b_1(b - t)}{12[bd - b_1(b - t)]}}$
	$bd - b_1(b - t)$	$x = \frac{2hb + h_1b_1}{2A}$ $n = b - x$	$\frac{2hb^3 + h_1b_1^3}{3} - Ax^2$	$\frac{I}{b - x}$	$\sqrt{\frac{I}{A}}$
	$cd + s(b - t)$	$n = \frac{d}{2}$	$\frac{sb^3 + s^3(b - t)}{12}$	$\frac{sb^3 + s^3(b - t)}{6d}$	$\sqrt{\frac{sb^3 + s^3(b - t)}{12[cd + s(b - t)]}}$
	$hs + kc$	$x = \frac{cr_1^2 + s^2(b - t)}{2A}$ $n = d - x$	$\frac{hs^3 + kc^3 - (b - t)(c - s)^3}{3} - (hs - t)(kc - s)^2$	$\frac{I}{d - x}$	$\sqrt{\frac{hs^3 + kc^3 - (b - t)(c - s)^3}{3(hs + kc - bs)} - (hs - t)(kc - s)^2}$
	$hs + kc + bs$	$x = \frac{sb^2 + s^2(b - t) + k(h - t)(2d - s)}{2A}$ $n = d - x$	$\frac{hs^3 + kc^3 + bs^3 - (b - t)(c - s)^3 - (h - t)(k - s)^3}{3} - (hs + kc + bs - 2A)x^2$	$\frac{I}{d - x}$	$\sqrt{\frac{I}{A}}$
	$hs + \frac{k(t + s)}{2}$	$x = \frac{2hs^2 + kh(2d + s) + h(h - t)(h + 3s)}{6A}$ $n = d - x$	$\frac{4hs^3 + h^3(3h + s)}{12} - Ax^2$	$\frac{I}{d - x}$	$\sqrt{\frac{I}{A}}$

frames of automobiles. In the chapter on frames, the width and height of the channel sections for various car frames are given. While the most advantageous form of beam, as far as strength is concerned, is the I-beam section, in frames the channel section is used, since it can be pressed into varying

heights, according to the strength required. We are referring to beams where the chief stresses are in the direction of the height of the beam. To obtain the maximum strength in a horizontal as well as a vertical direction, the square hollow section is the most efficient, while for stresses in every direction the round tube is the most efficient. It should be remembered that in vehicle frames, the material being steel, the ultimate strength is approximately the same in tension as in compression. If a structure, such as a channel section, for instance, is made of other material than steel, allowance must be made for the difference in its tensile and compressive strengths.

As an example, if a simple beam, carrying a certain load, is made of cast iron, the area of the lower flange, which is in tension, must be considerable larger than that of the upper flange, since

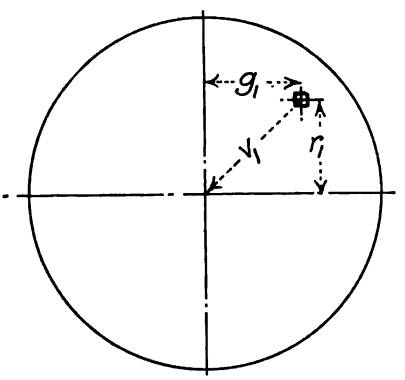


FIG. 27.—Illustrating polar moment of inertia.

cast iron has a very much lower strength in tension than in compression. The tensile strength of cast iron varies according to its grade, but it averages about 18,000 lb. per square inch, while its ultimate strength under compression is about 90,000 lb. per square inch.

The moment of inertia of a surface or an area should not be confounded with the moment of inertia applied to a body, as for instance, to the flywheel of an engine.

Polar Moment of Inertia.—The polar moment of inertia of a surface is the moment of inertia with respect to its axis through the center of gravity at right angles to the plane of the surface and it equals the sum of two moments of inertia taken with

respect to two gravity axes in the plane of the surface at right angles to each other.

Supposing in Fig. 27, a is a small elementary area at the distance r_1 from the horizontal neutral axis and q_1 distant from the vertical neutral axis. The polar moment of inertia J of the small area a is $= ar_1^2 + ag_1^2 = a\ (r_1^2 + g_1^2)$ and since $r_1^2 + g_1^2 = v_1^2$, the polar moment of inertia of the small area $= av_1^2$; if A represents the sum of all the elementary areas (as before, see page 43) and if r and q are the mean distances of all the said small areas from the horizontal and vertical axes respectively, the polar moment of inertia $J = Ar^2 + Ag^2 = A\ (r^2 + g^2) = Av^2$. The polar section modulus $Z_p = \frac{J}{v}$.

The polar moment of inertia and the polar section modulus are required when figuring torsion stresses in shafts as will be seen in Chap. V.

CHAPTER III

FRAMES

(FOR CALCULATING THE STRESSES IN FRAMES, SEE PAGE 77)

The frame of an automobile connects the various main units with each other and supports them in their proper places. Formerly, wooden frames were used to some extent, as wood deadens vibration considerably. Subsequently, wood, reinforced by steel plates, was employed, and finally steel frames were introduced. The ever increasing demand for rigidity and uniformity, has proven the pressed steel frame to be the most satisfactory. The objections to wood is that it cannot be made uniform, it is not weatherproof, and its source of supply is limited.

Side Members.—The frame of a motor vehicle comprises the usual channels, which run longitudinally at each side of the chassis and which are called side members or side rails, and the cross members which are attached to, and interconnect, the side members. As a rule, the frame members are made of sheet steel formed into channel sections; at times, in heavy trucks, some of the frame members are made of rolled channels. The cross members are mostly channel sections, sometimes U-sections, and occasionally tubular. Standard materials used for frames are open-hearth, hot-rolled, S.A.E. steel Nos. 1010, 1020, 1025, (see table IV). A number of motor vehicle manufacturers use heat treated frame members as they possess greater strength; yet, heat treatment involves extra cost and for this reason such frames are not always employed. In fact, some large frame producers do not furnish heat-treated frames at all, while others recommend them.

The frame is subjected to direct static stresses caused by the load resting on it, to jars and jolts transmitted to the frame through the spring; and to the continuous vibrations arising from the engine and transmission mechanism.

The strongest section of the frame should be at the point where there is the greatest bending moment. Where the bending moment decreases the side rails usually taper down, to save weight. A long wheel base, requires a stronger and heavier frame, for with

a given load the stresses are greater than in a shorter frame. The frame should be so designed as to secure maximum strength with lightness. There must be sufficient rigidity to prevent twisting or distortion of the frame, for distortion as a rule imparts strains to the bearings. (When one wheel rises over an obstruction, also when turning corners or applying the brakes suddenly the frame is subjected to great stresses.) Yet, too flexible a frame will put the bearings out of alignment and distort the body, and for this reason frames are now made considerably stronger than formerly. It is sometimes claimed that straight rolled channels are stronger and stiffer than pressed channels of variable section, due to the better disposition of the metal in the corners. However, a frame which is not too rigid has been found more satisfactory as the slight yielding (even if only very slight) under excessive shock, will prevent fracture or permanent set. Lightness is also sought, for extra weight means extra cost. A pressed steel frame is lighter than one made of rolled channels and the strength can be placed where required, but forming dies are needed for its production and these are expensive. A large number of dies are necessary for a frame, on account of the number of separate pieces of which it is composed; there are right and left side members, each requiring a separate die, in addition to those required for the cross members. The thickness of the frame metal varies from about $\frac{1}{8}$ in. up, for passenger cars, up to $\frac{3}{8}$ in. for heavy trucks, as can be noted from the illustrations in this chapter. Cross members are often made of thinner gauge stock than side members. Frequently, frames are narrowed in the front to permit the front wheels to sweep a great angle, in order to turn the car in a small radius.

Cross Members.—The location of the cross member in a frame is determined very largely by the location of the load, or the stresses on the frame. It would seem good practice to locate the cross members at or near the spring connections or spring brackets. Cross members are frequently used to support the clutch and brake lever actuating mechanism (with the unit power plant this is dispensed with), the transmission gear case, when placed amidship, the brake equalizing rods, etc. A great deal of care should be exercised in the design of the method of attachments between the cross members and the side members in order to save cost in labor and material and to obtain a more compact and rigid structure. When designing frames for "quantity production," the

quantity of material required is a very important factor, for in such cases material represents a far greater item than cost of labor. Sometimes it is found more economical to use separate gusset plates or small angles for joining frame sections, as this will save material even though requiring more labor. Many designers prefer the integral gusset while others prefer to rivet the top and bottom flanges of the cross member to those of the side member, bend over the web of the cross member and rivet this to the web of the side member. This method is called "bending ears."

Frame Brackets.—The frame proper is supported by the springs, through spring brackets, as shown in the drawings. The spring brackets are ordinarily steel castings, malleable castings, or drop forgings. Since the greatest concentrated forces on the side members arise from the reactions at the spring ends, the spring brackets must be so designed and attached to the frame, as to distribute the stresses arising from the spring ends over a considerable area.

The front of the frame is supported by semi-elliptic springs. The front spring hangers are placed inside of the curved front end of the side channels as seen from (Figs. 34 and 36). The rear end of the front spring is connected to the frame through spring shackles, whose object is to allow, for the increase in spring length which ensues when the latter is deflected, and for a decrease in spring length on the rebound.

In the rear of the side-rails, long brackets, named *goose-necks*, are often used, ordinarily these are steel forgings. With several types of rear springs, the goose-necks can be dispensed with. In trucks, where the frame overhangs the rear spring considerably, the rear ends of the springs (sometimes also the front ends) are attached to brackets which are connected together (from one side of the frame to the other) by a round steel bar, called spring bracket bar, to avoid twisting of the frame members. The rear springs are generally placed outside the frame, there being some clearance between the spring and the side members, hence the load and shocks imparted to the spring, exert a bending moment on the spring brackets which tends to twist the frame. The "spring bracket bar" effectively prevents this tendency as it locks the spring brackets together across the frame. It is good practice to place a cross member between the front spring hangers of the rear springs, or close to them, also at other points where the stresses are great.

A channel frame has a tendency to twist and buckle, and the cross members effectually stiffen it against these weaknesses.

With the Hotchkiss drive, the front hangers of the rear springs take the driving thrust, hence it is very important that these spring hangers should be securely attached to the frame and that the frame be here stiffened. In the Hotchkiss drive, the front eyes of the rear springs are connected to the frame by eyebolts and spring brackets, while the rear ends are shackled.

At the point where the frame is narrowed near the front, (see Figs. 39 and 44) there is an additional tendency for the frame to twist, hence this part is usually made stronger by providing an additional width to the top and bottom flanges, or by placing a cross member at this point or in close proximity thereto.

To prevent frame distortions where the radiator is located, and thus prevent damage to the latter, a cross member is placed underneath or close to it. Curves of any kind provided in side members will weaken them. The metal is also weakened when shaping the frame, and a small curve, or a small radius, will cause a greater change in the molecular structure of the metal.

Riveting.—The sizes of rivets used for frames varies from $\frac{5}{16}$ in. for small passenger cars, to $\frac{1}{2}$ in. for heavy trucks. The following illustrations show that a certain amount of clearance should be given the rivets in the holes. Riveting various members together while the rivets are red hot, gives the best satisfaction, since the rivets contract as they cool thereby pulling the metal surfaces together more firmly. It should be borne in mind that holes drilled in a frame will weaken it, when drilled, in or near, the upper or the lower flanges of the channels. The middle of the channel, at or near the neutral surface, can be drilled with impunity (see Chap. II). When holes are absolutely necessary in the upper or the lower flange, they should be as far distant as possible from the edge. Between the front and rear springs where the maximum strain occurs in the side members of a frame, the lower side of the channel is in tension, hence holes here have a more serious weakening effect than in the top flange; the latter is under compression and the rivets in the holes will help to resist the strain arising from the compressive stress in the metal. It is therefore more advantageous to attach brackets to the side web and the top flange.

Frame Design.—Figure 28 is the frame assembly of the U.S.A. Class B military truck, previously referred to. Attention is

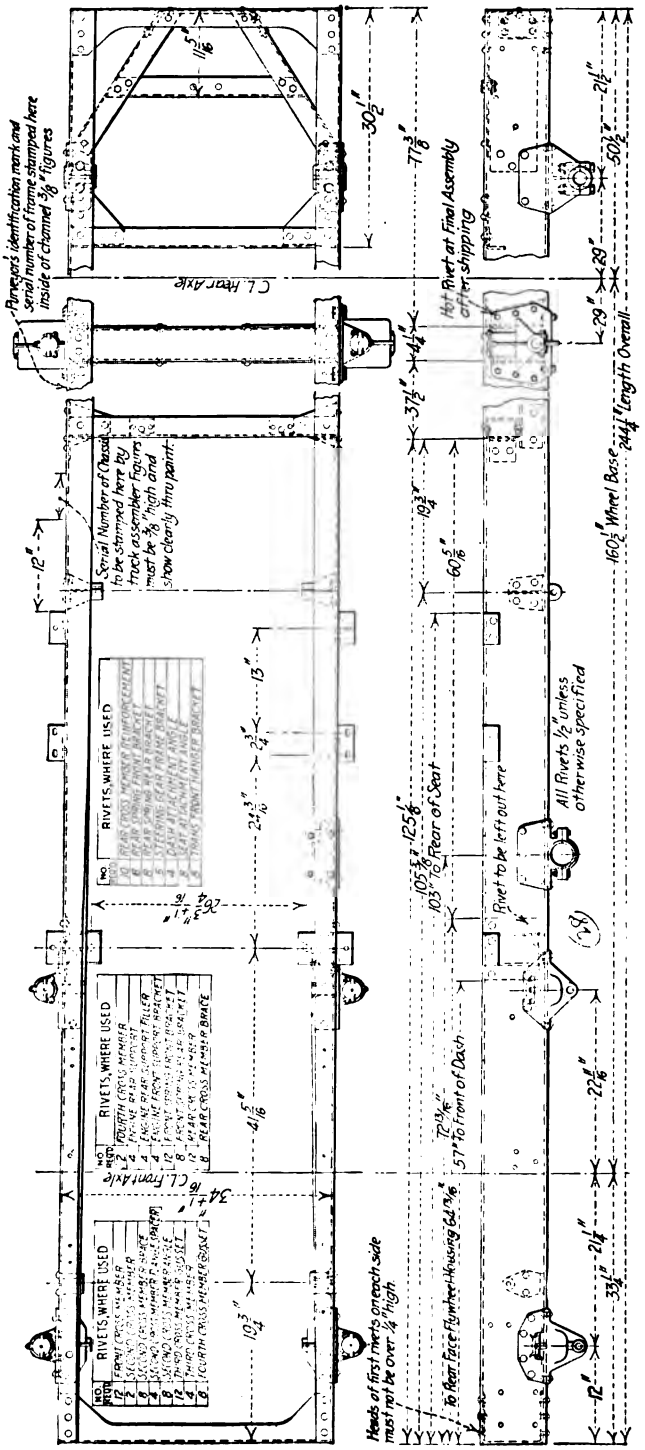

Parker's identification mark and serial number of frame stamped here inside of channel 3/8" figures
Serial Number of Chassis to be stamped here by truck assembler Figure must be 3/8" high and show clearly thru paint.
Hot Rivet at Final Assembly after Shipping
RIVETS, WHERE USED
REAR CROSS MEMBER REINFORCEMENT
REAR SPRING FRONT BRACKET
REAR SPRING REAR BRACKET
FRONT SPRING REAR BRACKET
FRONT SPRING FRONT BRACKET
REAR CROSS MEMBER
FRONT CROSS MEMBER
FRONT BUMPER BRACKET
FRONT CROSS MEMBER
FOURTH CROSS MEMBER
REAR CROSS MEMBER BRACE
C L Front Axle
C L Rear Axle
Heads of first rivets on each side must not be over 1/4" high
To Rear Face Flywheel Housing
57" to Front of Dash
Rear of Seat
Rivet to be left out here
All Rivets 1/2" unless otherwise specified
Wheel Base
Length Overall

Fig. 28.—Frame assembly of Class B truck.

called to the location of the brackets, some of which will be described, and to the notes on the top of the drawing showing the number of rivets required to attach the several parts to the frame members. All rivets are ½ in. unless otherwise specified.

"The frame is of open-hearth steel pressed into shape.[1] The side rails are channel section with parallel flanges which are tapered at the front to reduce weight and give greater room for the radiator. There is a heavy pressed steel cross member in front of the radiator, one between the front brackets of the rear springs, and one at the rear. A lighter pressed steel cross member supports the rear of the gear-box, while forged cross members support the front of the gear-box and the front of the engine. The frame is designed to be as flexible as possible without in any way being flimsy or likely to buckle.

"The pressed steel frame allows better distribution of metal than a rolled section, its uniform thickness is of advantage in riveting, there is better assurance of uniformity of metal, and the rounded portion where the flanges meet the web allows large fillets for brackets. The frame is somewhat narrower than usual, 34 in. to the outside edges of the side rails, in order to allow a great turning angle for the front wheels and to save weight in cross members and support brackets.

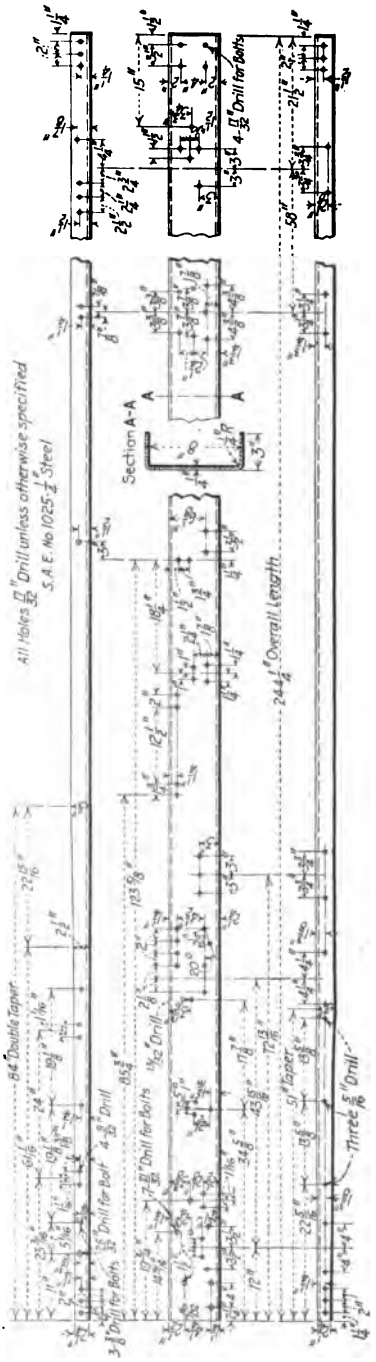


FIG. 29.—Side member (left hand) of Class B truck.

[1] See C. T. Myers, S. A. E. transaction, Part I, 1918.

"The wheelbase, named as 124 in. minimum and 156 in. maximum, in the Quartermaster General's office specifications, was increased to 160½ in. This was deemed advisable in order to keep the proportion of live load on the rear axle below 90 per cent, and at the same time have ample seat-width and sufficient space between the seat and the steering column."[1]

It might be mentioned here that the class B truck, nominally of 3½-ton capacity, is actually as strong as a 5-ton commercial truck.

Figure 29 shows the left frame side rail. The rivet holes are drilled $^{17}/_{32}$ in. in diameter, showing an allowance of $^{1}/_{32}$ in. between the size of the rivets and the holes. The center view of this figure is a side elevation showing the holes required at the sides of the frame. Above and below it are the plan view and the inverted plan view respectively, of this side member. The steel used is S.A.E. No. 1025. (For characteristics of this steel see Chap. XVII, Vol. I, Motor Vehicle Engineering, Engines.) It may be noted that most of the rivet holes are in the web (the vertical side) of the frame channel, and this is as it should be, since drilling holes through the top or the bottom flange weakens the structure as was mentioned before. There is no hole in the bottom flange at or near the middle of the chassis where the maximum strain in the side member occurs. The channel is 8 in. high and the top and bottom flanges are 3 in. wide.

Figure 30 shows the rear spring, rear bracket, and Fig. 31 the "rear spring shackle bar" which bridges the two rear spring brackets, from one side of the frame to the other, to give additional rigidity to the frame under load, as there is a tendency for the brackets to bend outwards and upward and this would twist the side members.

Figure 32 is the third cross member, showing a side elevation as well as the top and bottom views, with all holes marked thereon.

Figure 33 is the third cross member gusset (right rear and left front). These gusset plates are made of the same thickness as the frame channel, *i.e.*, ¼ in. Note that the radius of the bend is not less than the thickness of the material.

Figure 34 is the frame assembly of the four-cylinder Oldsmobile (45*A*), shown in Fig. 2*a*. The side members are made of $^{5}/_{32}$-in. stock and are interconnected by five cross members,

[1] See C. T. Myers, *S.A.E. Transactions*, Part 1, 1918.

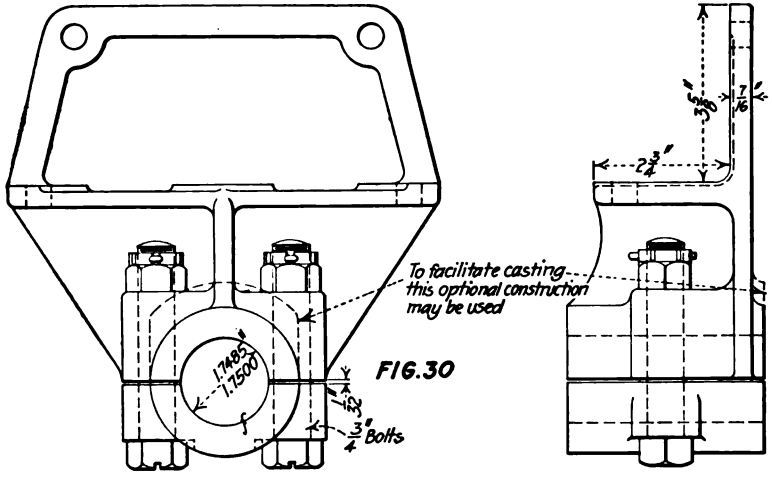


FIG. 30

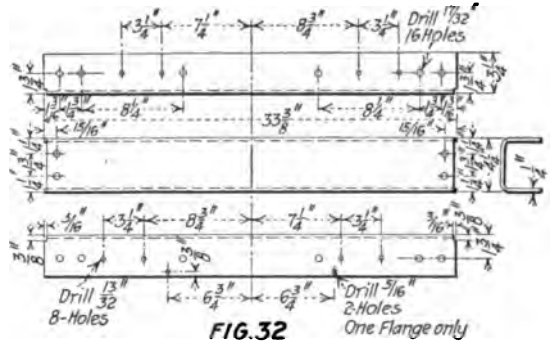


FIG. 32

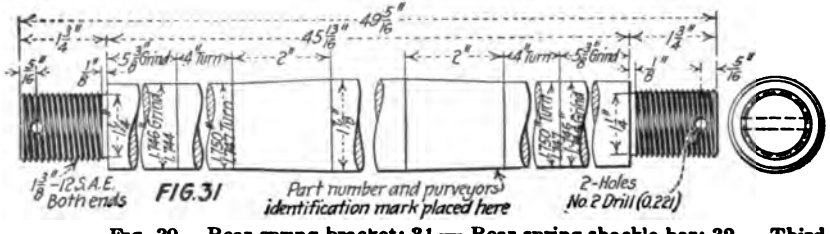


FIG. 31

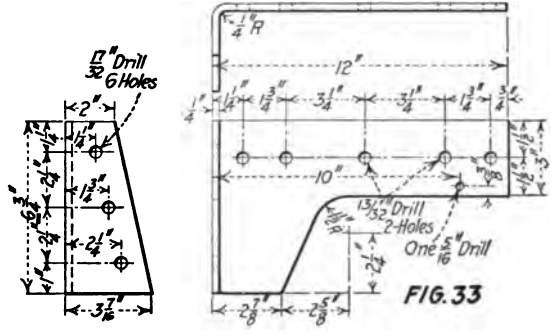
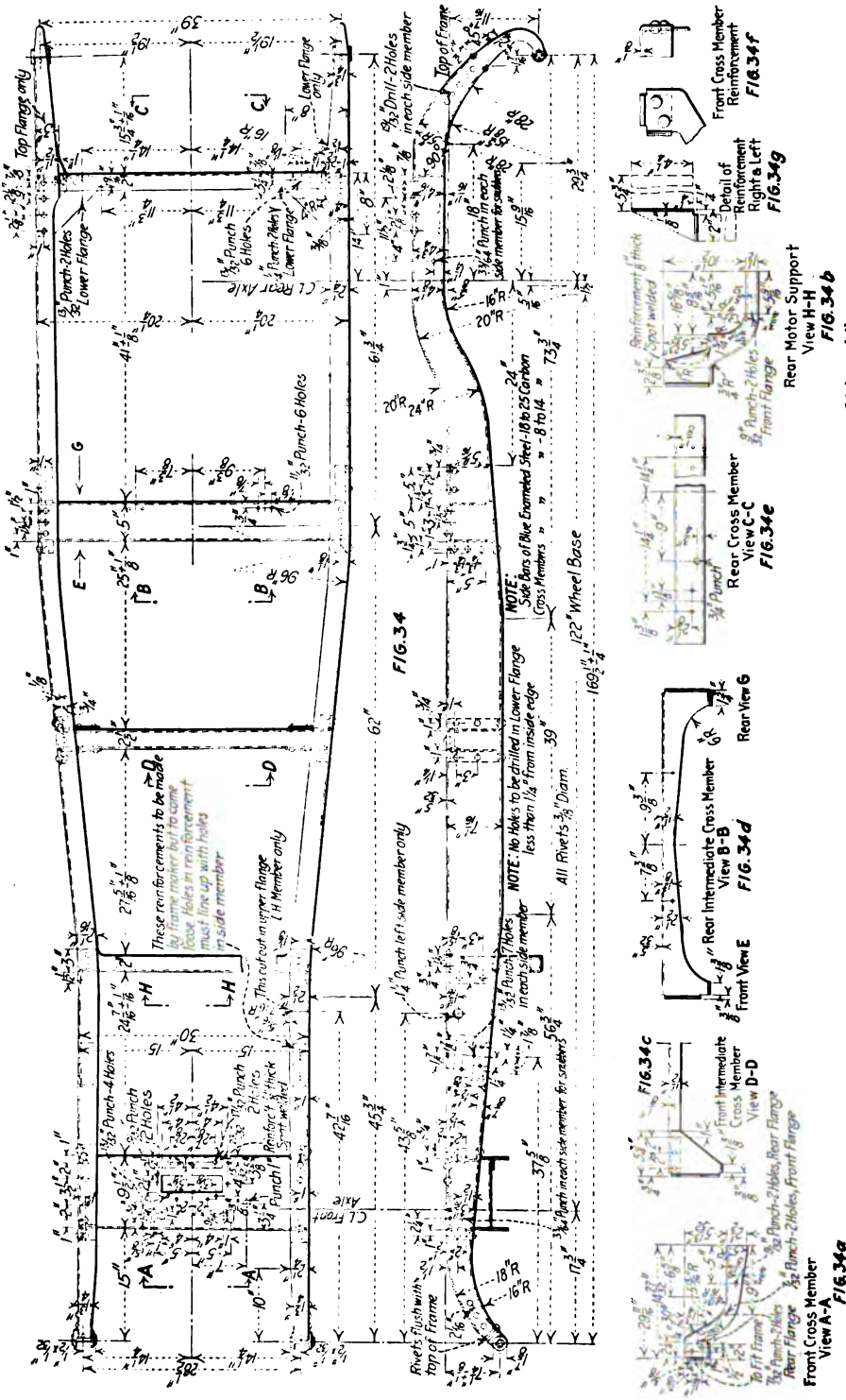


FIG. 33

FIG. 30.—Rear spring bracket; 31.— Rear spring shackle bar; 32.— Third cross member; 33.— Third cross member gusset.


Top Flange only
Lower Flange only
Punch 2 Holes Lower Flange
Punch 2 Holes Lower Flange
Punch 6 Holes
Punch-6 Holes
Drill-2 Holes in each side member
Top of Frame
Punch in each side member for gussets
Punch-4 Holes
Punch 2 Holes
Punch 2 Holes
Punch Material 3/8 Thick Scarf welded
These reinforcements to be made by frame maker but to come loose. Holes in reinforcement must line up with holes in side member
This cut-out in upper flange (LH Member only
Punch left side member only
NOTE: No Holes to be drilled in Lower Flange less than 1/4" from inside edge
Punch Holes in each side member
All Rivets 3/8" Diam.
NOTE: Side Bars of Blue Enameled Steel-10 to 25 Carbon Cross Members " " - 8 to 14 "
122" Wheel Base
Cl Front Axle
Cl Rear Axle
Rivets flush with top of Frame
Punch in each side member for gussets
Front Cross Member View A-A
Front Intermediate Cross Member View D-D
Rear Intermediate Cross Member View B-B
Front View E
Rear View G
Rear Cross Member View C-C
Detail of Reinforcement Right & Left
Front Cross Member Reinforcement
Rear Motor Support View H-H
Reinforcement Spot welded
Punch-2 Holes Front Flange
Punch-2 Holes, Rear Flange
Punch-2 Holes, Front Flange
To fit Frame
F16.34
F16.34a
F16.34b
F16.34c
F16.34d
F16.34e
F16.34f
F16.34g

FIGS. 34 to 34g.—Frame assembly and cross members of eight-cylinder Oldsmobile.

details of which are shown. Figure 34*a* is the front cross member which is located underneath the radiator and which also supports the front of the motor. It is 9½ in. wide and is made of ⅛-in. stock; 34*b* is the rear motor support, (2nd cross member), made of 5/32-in. stock; note the reinforcements in this and in the previous figure. Figure 34*c* is the 3rd cross member (also called front intermediate cross member); this cross member, as well as Fig. 34*d* (rear intermediate cross member) are U-sections, with the open side at the bottom; 34*e* is the rear cross member; 34*f* is an enlarged view of the front cross member reinforcement; and 34*g* are reinforcements for the inside of the side rails where the spring shackle brackets of the front springs are located. The details of the cross members are given; note how the front cross member is shaped to support the radiator and the front end of the motor. Note also the ribs extending at the bottom of this cross member (see Fig. 34) to give it sufficient strength. The rear motor support, 34*b* is shaped to receive the rear of the motor crankcase. Note the shape of the 4th cross member (rear intermediate) to obtain sufficient clearance for the motor shaft when the frame is in its lowest position, *i.e.*, under maximum spring deflection. It will be seen from the "Notes" that no holes are to be drilled in the lower flange less than 1¼ in. from the inside edge, that the side members are of 18- to 25-point carbon, and the cross members 8- to 14-point carbon steel. The drawings give the complete details of the frame as it is purchased from the frame makers.

Figure 35 shows a frame made by the Hydraulic Pressed Steel Co. for the Jordan car. This frame is suitable for a car weighing approximately 3,000 lb., it has a wheel-base of 127 in. The top view shows that the side rails run in a straight line from the rear where the frame width is 43¼ in., outside dimension, to 27 13/64 in. in the front. Wherever the cross members are rivetted to the side rails, the flanges of the latter are widened as shown at *X*. The side rail is made deep in the middle (6¼ by 2 5/32 in.) where there is the greatest bending moment, gradually tapering towards the ends. The front center cross member (*AC* 988), is an angle 1¾ by 2½ by ⅛ in. thick. The rear center cross member (*AC* 989) is of channel section (3½ by 1¾ by ⅛ in.) but it is attached to the frame as a U-channel. The rivets used are 5/16 in. for attaching the frame members together, the holes being 1 1/32 in.

In some frames the side members are made of chrome-nickel

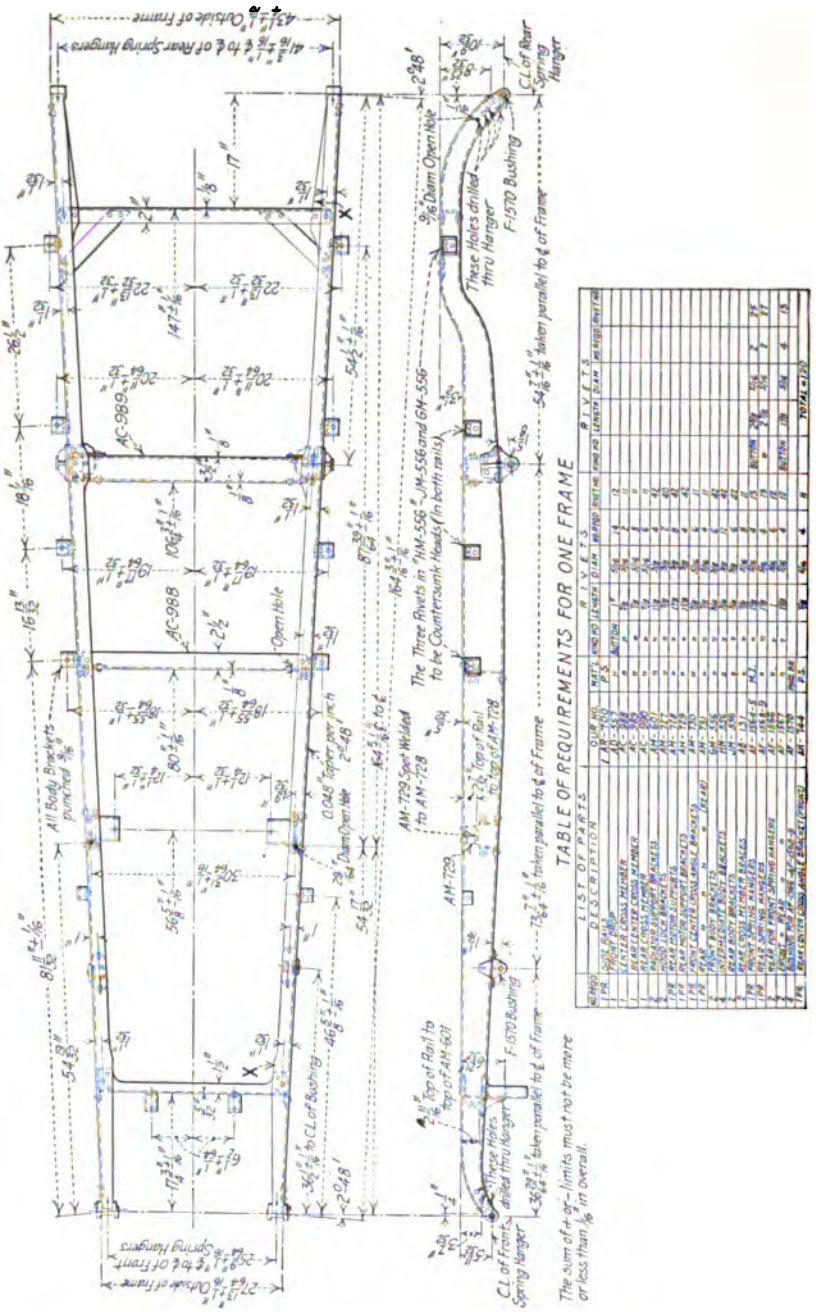


TABLE OF REQUIREMENTS FOR ONE FRAME
LIST OF PARTS
RIVETS
All Body Brackets punched 5/8"
Open Hole
The Three Rivets in "HM-556" JM-536 and GM-536 to be Countersunk Heads (In both rails)
AM-729 Spot Welded to AM-728
These Holes drilled thru Hanger
F-1570 Bushing
C.L. of Rear Spring Hanger
C.L. of Front Spring Hanger
The sum of ar- limits must not be more or less than 1/16" in overall.

FIG. 35.—Frame (made by Hydraulic Pressed Steel Co.) of Jordan car.

TABLE IV.—STEEL STANDARD

Parish Manufacturing Co.

Chemical properties									Physical properties				
Name	Grade No.	Carbon	Manganese	Phosphorus	Sulphur	Nickel	Chromium	Vanadium	Elastic limit, pounds per square inch	Ultimate strain, pounds per square inch	Elongation		Remarks
											Inches in 2 in.	Per cent of length	
Chrome-vanadium	6	.22–.28	.60–.80	.03 Max.	.04 Max.	.....	.80–1.00	.15 Max.	96,150	121,100	.375	18.75	Heat treated
Chrome-nickel	1	.20–.26	.45–.55	.03 Max.	.04 Max.	1.40–1.60	.45– .60	.....	88,700	104,400	.38	19.0	Heat treated
Carbon	3	.20–.28	.45–.55	.03 Max.	.04 Max.	.....	.....	.....	67,150	92,200	.42	21.0	Heat treated
Carbon	3	.20–.28	.45–.55	.03 Max.	.04 Max.	.....	.....	.....	41,200	62,100	.67	33.5	Untreated
Carbon	4	15–.19	.40–.45	.03 Max.	.04 Max.	.....	.....	.....	34,300	61,100	.68	34.0	Untreated
Carbon	5	.08–.14	.40–.45	.03 Max.	.04 Max.	.....	.....	.....	34,000	53,000	.74	37.0	Untreated

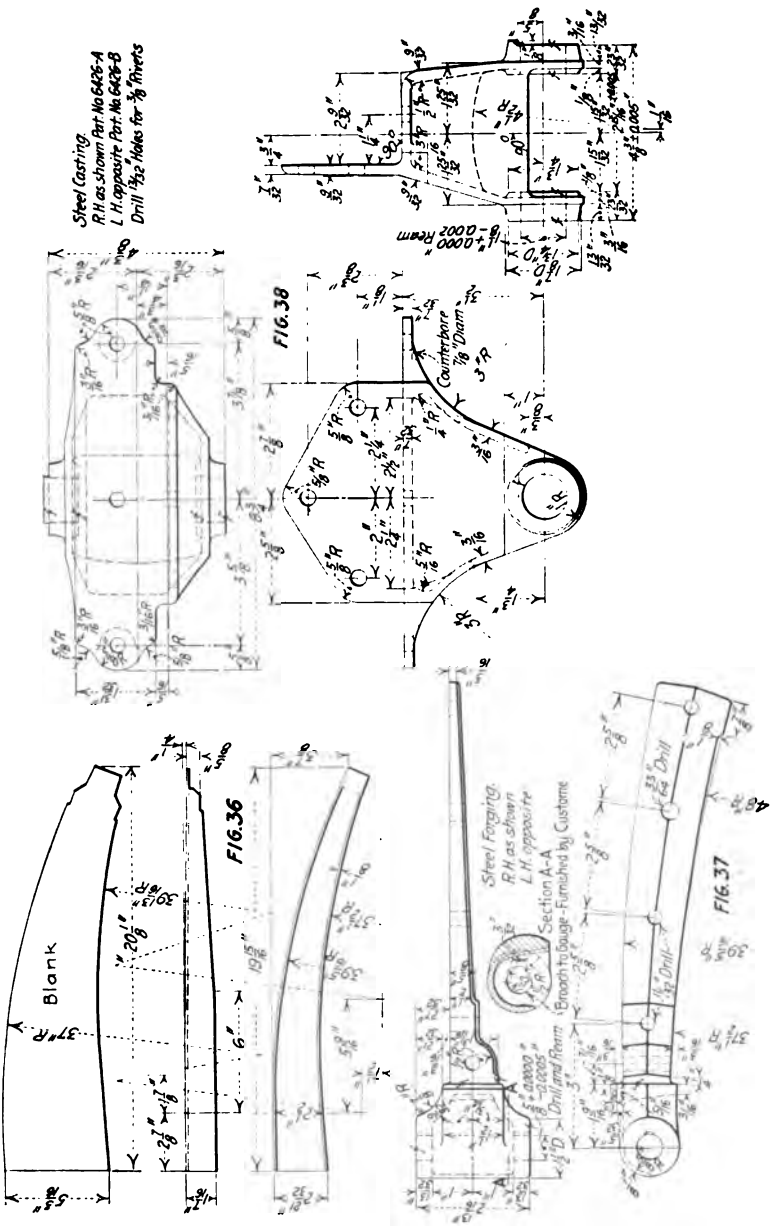


FIG. 38.—Cover plate for front spring hanger. FIG. 37. FIG. 38.—Front and Rear spring brackets for front spring (Pierce-Arrow passenger car).

steel, heat treated, while the cross members are of low carbon steel, untreated. The accompanying table gives the standard steels used by the Parish Manufacturing Co. and their physical properties. Four samples were tested of each steel and the results noted as shown.

When there are sharp curves in cross members, they are stamped hot, but simple stampings are ordinarily stamped cold.

Figure 36 shows the front cover plate which covers the inner side of the front of the frame where the front spring hanger (Fig. 37) is located, as used on the large Pierce-Arrow passenger car.

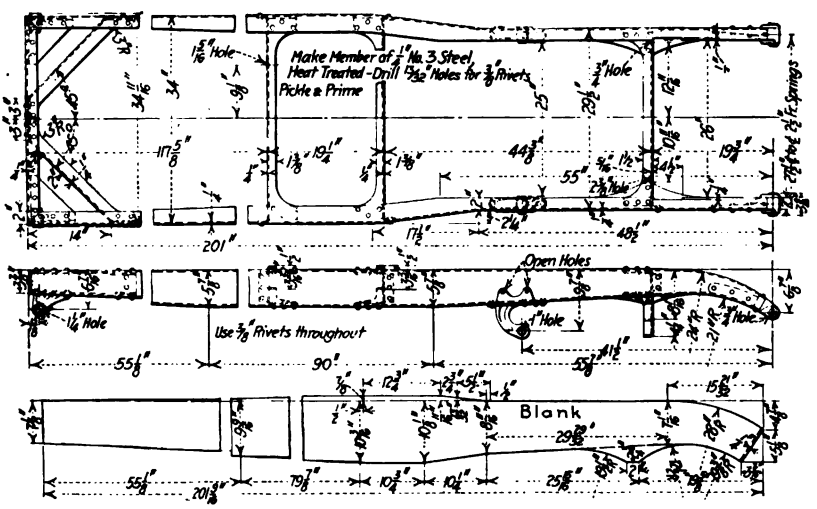

Make Member of 1/4" No. 3 Steel, Heat Treated - Drill 5/32" Holes for 3/8" Rivets, Pickle & Prime
Open Holes
Use 3/8" Rivets throughout
Blank
201"
90"

FIG. 39.—Frame (made by Parish Mfg. Co.) of White two-ton truck.

Figure 38 gives the detail of the rear spring bracket of the front springs; this is a steel casting.

Figure 39 is the general frame assembly of the White 2-ton truck, (frame made by the Parish Mfg. Co.). The total length of this frame is 201 in., and the greatest depth of the channel is 5⅞ in. The lowest view gives the blank, of which the side member is stamped. Figures 40*a* and *b* are detail drawings of the front, and third cross members respectively. The second cross member is the same as the third, except that its position is reversed as shown in the assembly drawing. Particular attention is called to Fig. 40*a*, showing how the ends of the member is formed and where the holes are placed for riveting it to the side members.

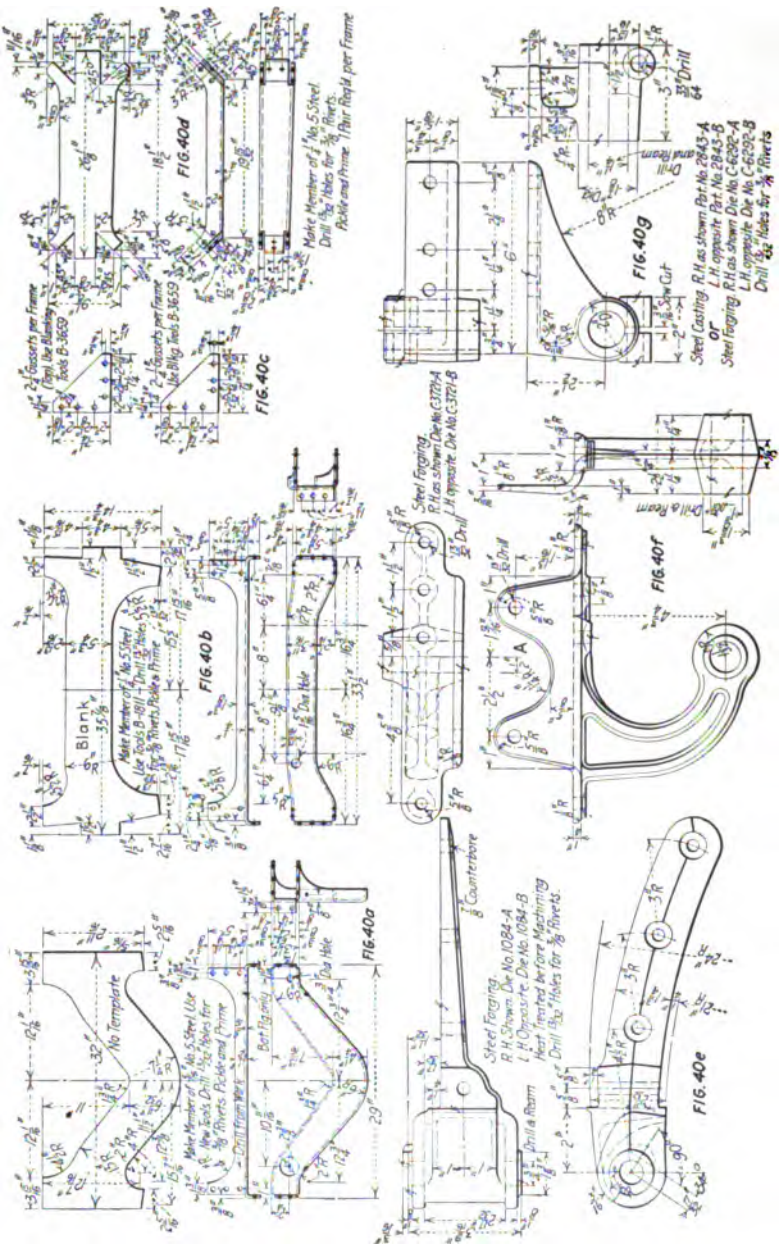

F16.40a
F16.40b
F16.40c
F16.40d
F16.40e
F16.40f
F16.40g
No Template
Blank
Make Member of 1 No.5 Steel Use Tools B-811 Drill 13/16 Holes 5 R
Make Member of 1 No.5 Steel Use Tools B-811 Drill 13/16 Holes for 3/4" Rivets Pickle and Prime
Make Member of 1 No.5 Steel Use Tools B-811 Drill 13/16 Holes for 3/4" Rivets Pickle and Prime Drill from Web
2 Gussets per Frame Use Bldg Tools B-3659
2 Gussets per Frame Use Bldg Tools B-3659
Make Member of 1 No.5 Steel Drill 13/16 Holes for 3/4" Rivets Pickle and Prime 1 Pair Reqd per Frame
Steel Forging R.H. Shown Die No.1084-A L.H. Opposite Die No.1084-B Heat Treated before Machining Drill 13/16 Holes for 3/4" Rivets
Steel Forging R.H. as shown Die No.C3721-A L.H. opposite Die No.C3721-B
Steel Casting R.H. as shown Part No.2843-A L.H. opposite Part No.2843-B
or
Steel Forging R.H. as shown Die No.C-6292-A L.H. opposite Die No.C-6293-B
Drill 13/16 Holes for 3/4" Rivets
Drill
Ream
Counterbore
Saw Cut

FIG. 40.—Cross members and brackets made by Parish Mfg. Co. (White two-ton truck frame). *a*, Front cross member; *b*, third cross member; *c*, gusset; *d*, brace; *e*, front spring rear bracket; *f*, front spring hanger; *g*, rear spring hanger.

The side members are made of ¼-in. stock, carbon steel, heat treated. The second, third and rear cross members are of carbon steel untreated, ¼ in. thick, while the front cross member is of 5/16-in. stock, carbon steel, untreated.

Figure 40*c* also shows the gussets which attach the rear cross member to its side members.

Figure 40*d* is the diagonal brace, in the rear, which has a considerable stiffening effect. All the rivets in the frame are ⅜ in. in diameter.

Figure 40*e* is the front spring hanger, which is a steel forging.

Figure 40*f* is the front spring bracket to which are attached the shackles for connecting it with the rear end of the front spring. Attention is called to the I-section of this forging to obtain maximum strength with minimum weight, the center web of the I-section being only ¼ in. thick. Note how weight is saved in the vertical flange, at *A*. The overall length of this bracket where it is attached to the side member is quite considerable, in order to distribute the pressure over a larger area of the frame member, and also because the stresses are not always purely vertical. When the spring is flexing it will impart forces against this bracket in a forward and backward direction, depending on the angle of the shackles with respect to the frame.

Figure 40*g* is the rear spring hanger attached to the rear of the frame. This hanger is sometimes a steel casting, at other times a steel forging. A steel forging is, of course, stronger and can be manufactured somewhat lighter for a given strength, but unless it is to be used in sufficiently large quantities, it is not economical to use forging dies, since they are very expensive. Patterns for the foundry, on the other hand, are comparatively inexpensive.

The following side members (often called side rails) are examples of standard designs, supplied by the Savage Arms Corporation.

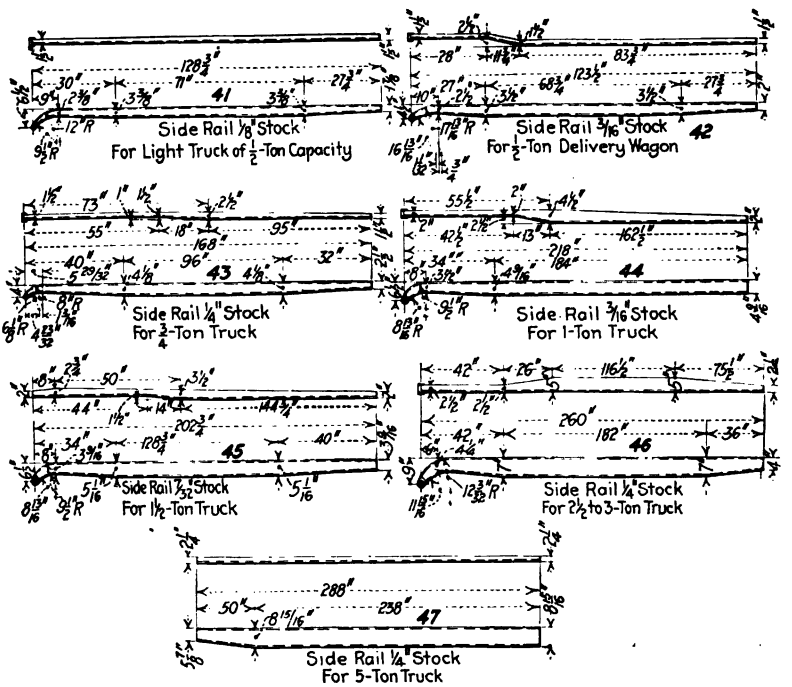
Figure 41 is a straight side rail for a light delivery truck frame of ½-ton capacity.

Figure 42 shows a side member with a sharp compound bend. It is intended for a delivery wagon of the same capacity as the frame of Fig. 41.

Figures 43 and 44 show the frame strengthened at the compound bend to compensate for the weakening effect which invariably accompanies such bends and to give additional strength

in the middle. Bends of any kind render the frame more liable to twisting strains.

Figure 43 is suitable for a ¾-ton truck frame; it has an overall length of 168 in. The height and thickness of stock are shown; the width of the flange is 2½ in. and as was stated in Chap. II on beams, the quantity of material at the greatest distance from the neutral surface determines the strength of a beam.

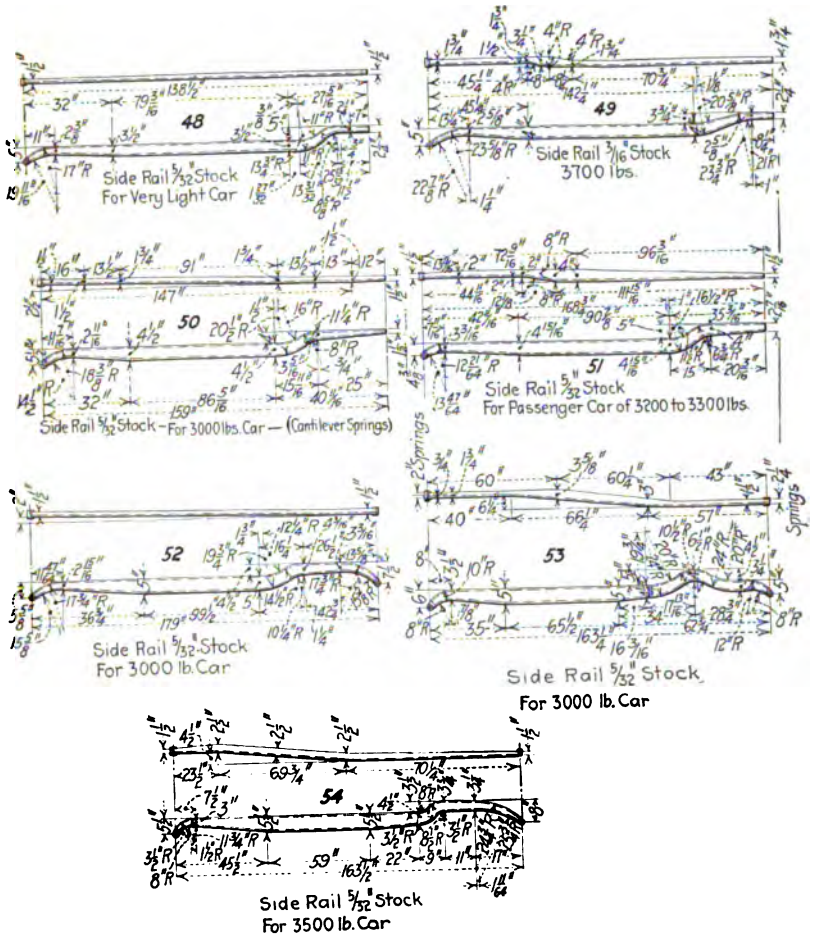


FIGS. 41 to 47.—Side members for truck frames manufactured by the Savage Arms Corp.

Figure 44 is suitable for a 1-ton truck. Figure 45 shows a side rail for a 1½-ton truck. It should be noted that the compound bend is always located where the frame is comparatively deep; it is not good practice to have a compound bend where the frame is made shallow on account of its weakening effect.

Figure 46 shows a more unusual type of frame. The width of the top and bottom flanges are increased where the stresses usually are great in order to properly apportion the strength to the load. It is suitable for a truck of from 2½- to 3-tons capacity.

Frequently truck frames have side rails, almost, or entirely, straight, and of the same dimensions throughout their entire length, as shown in Fig. 47, which is suitable for a 5-ton truck. In such designs there is necessarily more metal in some portions of the frame than required but much simpler dies can be used.



FIGS. 48 to 54.—Side members of passenger car frames, manufactured by the Savage Arms Corp.

Unless a large number of frames are made the die cost is a very important item.

Figures 48, 49, 50, 51, 52, 53 and 54, show a variety of designs where a "kick-up" is provided, *i.e.*, the rear end of the frame is raised in order to clear the axle. Figure 48 is suitable for a very

light passenger car, weighing not more than 1,500 or 1,800 lb., as for instance, cars like the Ford, or Chevrolet 490, although as a rule, deeper channels are used for such long frames. Figure 49 is made lighter than is common practice today; usually frames of such a length are made deeper than 4 in. These two frames are intended for use with either cantilever or three-quarter elliptic springs, and the frame of Fig. 49 is suitable for a car weighing about 3,700 lb. Figure 50 is a good design for a car weighing about 3,000 lb. and equipped with cantilever springs. Note the difference in size of this frame compared with the last two which are too light, according to present-day practice.

Figure 51 is suitable for a passenger car weighing from 3,200 to 3,300 lb.

Figures 52 and 53 are suitable for passenger cars of about 3,000 lb., equipped with semi-elliptic springs.

Figure 54 is suitable for passenger cars of about 3,400 or 3,500 lb., equipped with semi-elliptic springs.

Frequently, goosenecks are provided in the rear for attachment of the springs.

Figures 55 to 74 inclusive show different designs of cross members used in frames, manufactured by the Savage Arms Corporation.

Figures 55, 56, 57, and 58 are cross members for supporting the front of the motor and the radiator.

Figures 59 and 60 are cross members used to support subframes for engine mounting.

Figures 61, 62, 63 and 64 are front cross members.

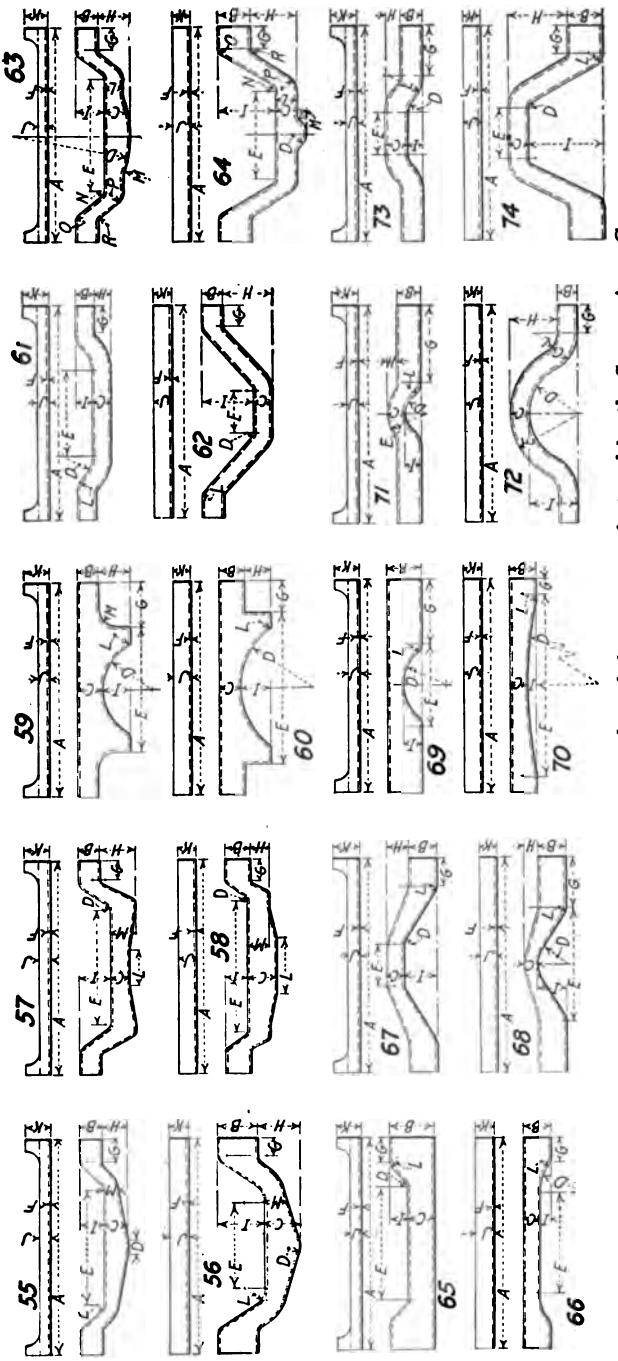
Figures 65 and 66 are center cross members, used for suspending the transmission or supporting subframes.

Figures 67 and 68 are center cross members suitable for carrying the propeller shaft bearings, or for supporting transmissions. Sometimes also for supporting subframes.

Figures 69, 70, 71, 72, 73 and 74 are cross members with a "kick-up" to clear the propeller shaft.

The material for frames used by the Savage Arms Corporation, is open hearth steel adapted for cold pressed frame construction. The strength of the material is approximately as follows:

Ultimate tensile strength	Over 58,000 lb. per square inch
Elastic limit	Over 36,000 lb. per square inch
Elongation in inches	Over 35 per cent
Reduction in area	Over 50 per cent



FIGS. 55 to 74.—Cross members for car and truck frames, manufactured by the Savage Arms Corp.

In the following table, the letters refer to those on the drawings and it shows the various dimensions of similar types of cross members.

TABLE V

FIG. NO.	A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	R
55	34	2 3/4	3 1/4	2	20	3/16	2 3/4	3 13/16	3 5/16	2	6	1	2				
"	36	4	4 1/2		20 1/4	1/4	4 1/2	3 1/2	3	2 1/2	4 1/2	5/8	3 3/4				
"	40	5 13/32	6		22	1/4	4 1/2	4 3/32	3 1/2	3 1/4	4 1/2	1	5 1/8				
56	38	3	3		24	3/16	3	3 1/8	3 1/8	3	5	1	2 1/2				
"	42	5 5/8	6 1/2	3 1/4	16	1/4	4	7 19/32	6 25/32	2	4	6 1/2	4 3/4				
"	34	2 1/32	2 5/8	4 1/4	17	5/32	3 1/4	4 3/16	3 13/32	2	5 1/2	3	1 29/32				
"	36	3 1/8	3 3/4	1 11/16	19	3/16	3 1/2	6 3/8	5 3/4	2	4 1/2	2 1/4	3 1/2				
57	36	3 1/8	2 11/16	3/4	18 5/16	5/32	4 1/2	4 1/2	4 1/4	2		5	3 3/8				
58	36	3 1/4	4 1/4	1/2	20 1/2	3/16	5	4 1/8	3 1/8	2	4 1/2	8	3 7/8				
59	36	2 3/4	3 1/4	7	17 5/16	5/32	9 11/32	4 17/32	4 1/32	1 1/2		1	2 1/2				
60	36	3 5/8	3	10 3/8	21 1/8	3/16	7 7/16	3 5/8	4 1/4	1 1/2		1					
"	36	2 13/16	3	10 3/8	21 1/8	3/16	7 7/16	4 7/16	4 1/4	1 1/2		1					
61	36	4 5/8	2 1/2	4 1/4	18 1/2	3/16	2 1/2	2	4 1/8	1 3/4	6	1					
"	32	2 11/32	2 1/4	4	6	3/16	5 3/4	1 9/32	1 3/4	1 1/2	4	6					
"	36	2 25/64	2 1/2	3/4	21	5/32	4 1/2	2 47/64	2 5/8	1 3/4	5	3/4					
"	38 1/2	4 3/16	2 1/4	8		5/32	5 10/32	5 13/16	7 3/4	1 3/4	4 1/2	3					
62	40	6 1/2	3 1/4	3/4	10	1/4	3 1/2	8	11 1/4	2	6	3/4					
"	36	3 3/16	2 1/4	1	10	5/32	3 29/32	6 1/4	7 3/16	1 3/4	4 1/2	4 3/16					
"	36	4 1/4	3	2	16 1/4	1/4	3 5/8	3 1/2	4 3/4	2	6 1/2	1 1/2					
63	36	3 1/16	4 3/4	12	16 7/8	3/16	4 1/2	3 7/8	2 3/16	2	6		20	5	1 7/8		
"	36	3 9/32	4 1/16	16	20 3/4	5/32	4 7/16	4 1/32	2 5/8	2	5 1/2	2 9/16	1 3/8	1	1 1/2	2 3/4	5/8
64	38	3 7/16	8	2 5/8	16	3/16	5 11/16	2 13/16	1 1/2	2		4 3/4	3 1/2	4 3/4	4 3/4	2 1/16	1 1/2
"	36	3 37/64	4 1/2	4	19 3/4	3/16	4 5/8	3 35/64	2 5/8	1 1/2	4 1/4	2 7/8	4	1	1 1/16	3 7/8	11/32
"	43	4 23/32	6	3	13 1/2	1/4	5 1/2	5 25/32	6	2	5	4 1/2	3	9 5/8	1	8 1/2	1 1/2
65	36	6 1/4	3 1/16	1	20	1/4	4 1/2		2 9/16	3	6 1/2	1					
"	42	8 7/16	6 1/4	4	10	1/4	10 1/16		2 3/16	2	4	4					
"	38	4 1/8	3	1 1/2	10	3/16	4 3/4		1 7/8	2 1/2	6	1 1/2					
"	40	8 3/16	3 1/8	2	16	1/4	5 1/4		5 1/16	3		2					
"	38	8 7/16	5 5/16	1 1/4	15	1/4	8 1/4		3 1/4	2	10	1 1/4					
66	38	3	2 3/16	2 1/4	22 1/2	3/16	5 1/2		13/16	1 1/2	3 3/4	3/4					
"	36	4 1/2	3 3/4	4	20	3/4	4 1/2		3/4	3	4	4					
67	36	2 3/4	2 1/4		18	5/32	5	13/32	2 9/32	1 1/2	2 1/4						
"	40	5 3/4	4 1/4	3/4	7 1/2	1/4	11 5/8	2 1/4	3 3/4	1 3/4	4 1/2	3/4					
"	40	8 7/16	5	1	19 1/2	1/4	6 3/4	2 3/8	5 13/16	2	4 1/2	1 1/2					
"	36	5 1/4	3 1/2	1 3/8	14	3/16	4	2 15/64	4 3/64	1 3/4	4 1/4	1 1/2					
"	34	2 13/16	2 1/4	2	11	5/32	6	7/8	1 7/16	1 1/2		2					
68	42	3 3/16	2 7/16	22 3/4	25 5/8	5/32	8 3/16	3/4	1 1/2	1 1/2							
"	38	4 3/16	2 3/4	3 1/2	26 5/8	5/32	5 11/16	1 3/16	2	2	8	1					
69	36	3 5/64	2 1/4	4	8 1/2	1/4	3 3/4		2 1/4	1 3/4	4 1/2	1					
"	34	3 7/16	2 13/16	6	5 3/4	1/8	14 1/8		5/8	1 1/2	5 1/2	1 1/8					
"	40	6 1/2	4	4	9 3/4	1/4	15 1/8		2 1/2	3	6 1/2	2					
"	38	7 1/2	6	4	7 1/4	1/4	15 3/8		1 1/2	2		1 1/4					
70	38	7 1/2	6	24	30	1/4	4		1 1/2	2							
71	34	3 7/16	2 1/4	2 15/16	5 3/16	3/16	13	13/32	2 1/32	1 1/2	3 3/4	2 1/4					
"	36	4 3/16	3	5	8	5/32	4 1/2	6 13/16	8	2	6	2					
"	38	4 5/8	3 5/8	3	6 5/8	5/16	4	3 5/8	4 5/8	2	6	8					
"	36	2 1/4	1 3/4	4	6 1/4	1/8	11	2 1/8	3 1/8	2	3 3/4	5/4					
72	40	2 9/16	2 9/16	8	10 9/16	5/32	8 1/4	6 1/2	6 1/2	2 1/2		3					
"	36	3	3	6	9	1/8	9 1/4	3 27/32	3 27/32	2	3 1/4	2					
73	44	4 11/16	3	4	26	5/32	4	1 5/32	2 27/32	1 1/2		2					
"	36	4 5/8	2 1/2	1 11/32	14	3/16	4	1 19/64	3 27/64	1 1/2	5	1 11/32					
"	38	3	3	2	15 1/2	1/8	7 21/32	1	1	1 1/2		5					
"	40	5 1/2	3 5/8	3/4	7 1/2	1/4	11 5/8	1	2 7/8	1 3/4	4 1/2	3/4					
74	40	5 11/16	2 5/8	2	18 1/2	3/16	4	4 3/4	7 3/16	1 1/4	4	2					
"	36	4 11/32	2 1/2	4 1/4	8 1/4	5/32	4 1/4	6 29/32	8 3/4	1 3/4		1 1/2					
FIG. NO.	A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	R

Figure 75 shows the assembly of the frame used in the Daniels motor car, (manufactured by the Parish Mfg. Co.) which is a

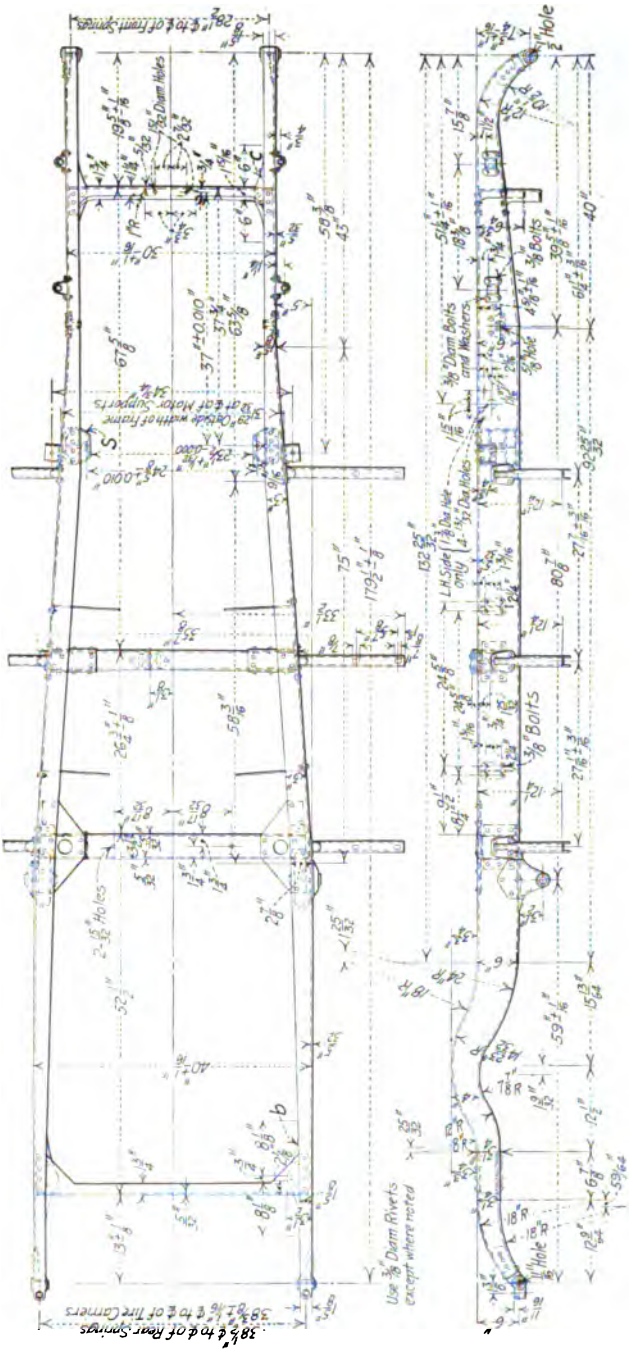


FIG. 75.—Frame made by Parish Mfg. Co. for Daniels car.

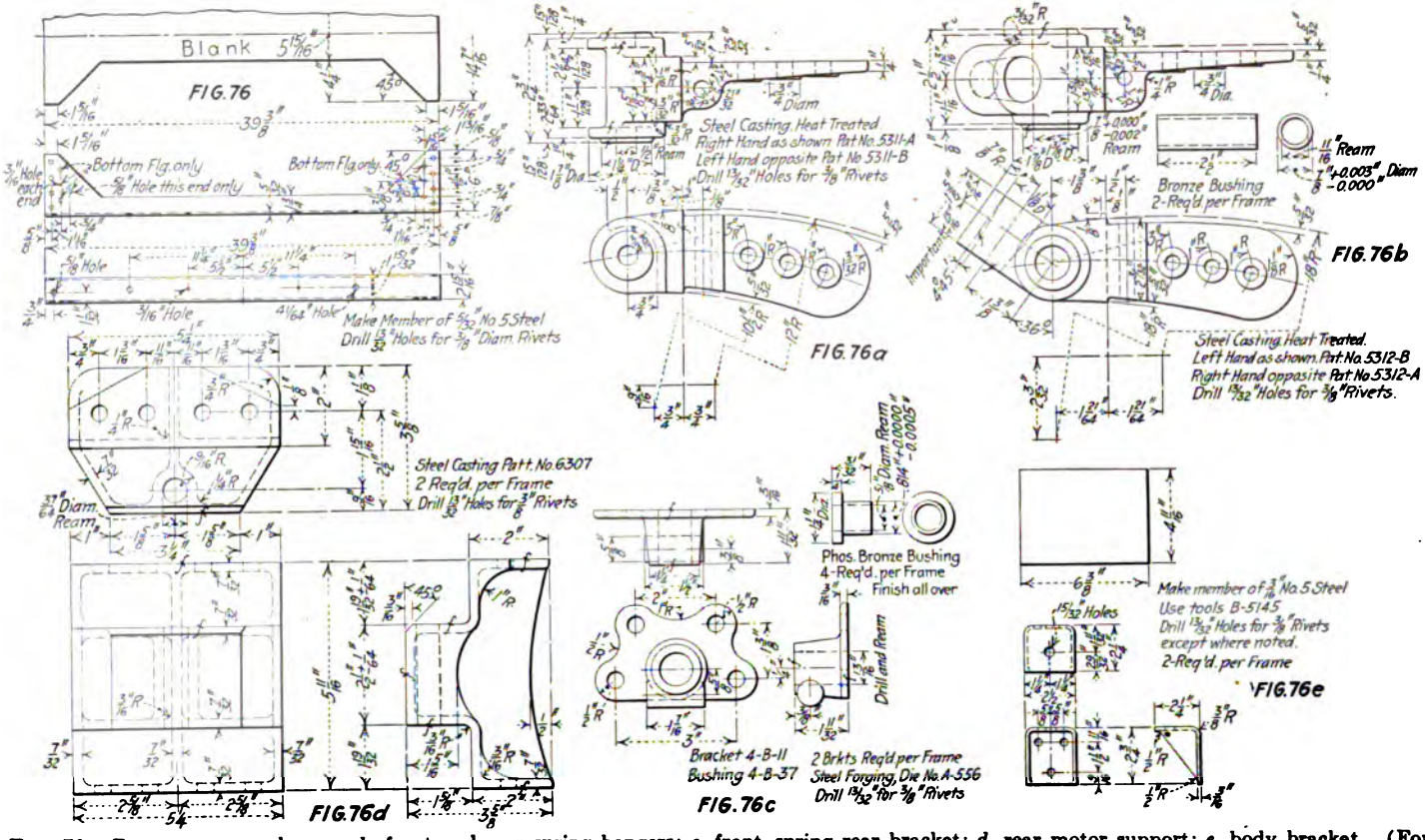

Blank
FIG. 76
Bottom Flg. only
Hole this end only
Make Member of 5/32" No. 5 Steel
Drill 13/32" Holes for 3/8" Diam. Rivets
FIG. 76a
Steel Casting, Heat Treated
Right Hand as shown Pat No. 5311-A
Left Hand opposite Pat No. 5311-B
Drill 13/32" Holes for 3/8" Rivets
Bronze Bushing
2-Req'd per Frame
FIG. 76b
Steel Casting Heat Treated.
Left Hand as shown Pat No. 5312-B
Right Hand opposite Pat No. 5312-A
Drill 13/32" Holes for 3/8" Rivets.
Steel Casting Patt. No. 6307
2 Req'd. per Frame
Drill 13/32" Holes for 3/8" Rivets
FIG. 76d
Phos. Bronze Bushing
4-Req'd. per Frame
Finish all over
Bracket 4-B-11
Bushing 4-B-37
2 Brkts Req'd per Frame
Steel Forging, Die No. A-556
Drill 13/32" for 3/8" Rivets
FIG. 76c
Make member of 3/16" No. 5 Steel
Use tools B-5145
Drill 13/32" Holes for 3/8" Rivets
except where noted.
2-Req'd. per Frame
FIG. 76e

FIG. 76.—Rear cross members; *a*, *b*, front and rear spring hangers; *c*, front spring rear bracket; *d*, rear motor support; *e*, body bracket. (For Daniels car frame.)

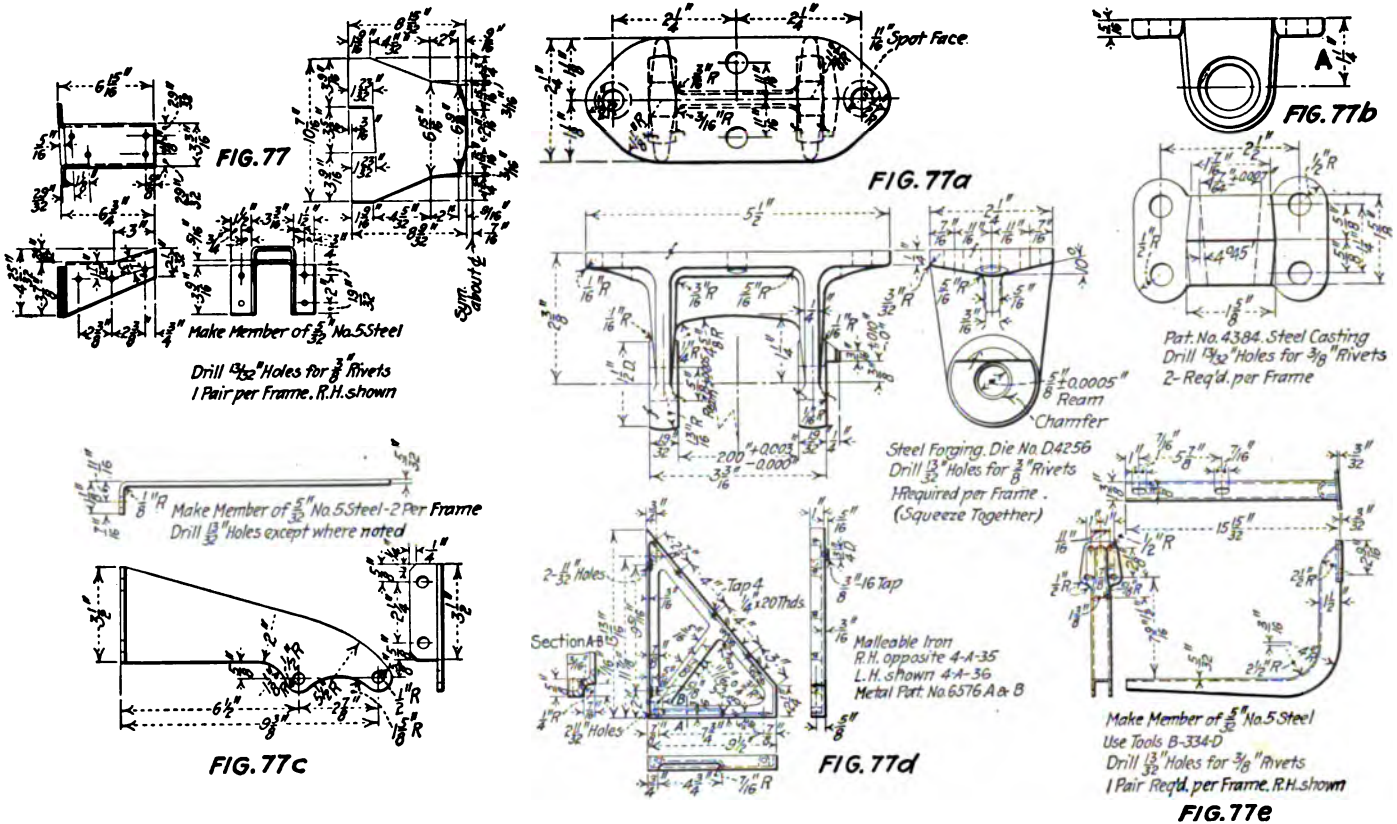


FIG. 77
Make Member of 5/32 No. 5 Steel
Drill 13/32" Holes for 3/8" Rivets
1 Pair per Frame. R.H. shown
FIG. 77a
11/16" Spot Face
FIG. 77b
Pat. No. 4384. Steel Casting
Drill 13/32" Holes for 3/8" Rivets
2- Req'd. per Frame
Steel Forging. Die No. D4256
Drill 13/32" Holes for 3/8" Rivets
1-Required per Frame.
(Squeeze Together)
5/8 +0.0005" Ream
Chamfer
FIG. 77c
Make Member of 5/32" No. 5 Steel-2 Per Frame
Drill 13/32" Holes except where noted
FIG. 77d
Section A-B
2-11/32" Holes
4-Tap 1/4 x 20 Thds.
3/8-16 Tap
Malleable Iron
R.H. opposite 4-A-35
L.H. shown 4-A-36
Metal Part No. 6576 A & B
FIG. 77e
Make Member of 5/32" No. 5 Steel
Use Tools B-334-D
Drill 13/32" Holes for 3/8" Rivets
1 Pair Req'd. per Frame. R.H. shown

FIG. 77.—Gusset; *a*, torque arm bracket; *b*, fender socket; *c*, muffler support; *d*, toe board bracket; *e*, step bracket. (For Daniels car frame, manufactured by Parish Mfg. Co.)

high priced car, having an eight-cylinder motor with bore and stroke of $3\frac{1}{2} \times 5\frac{1}{4}$ in. Notice at *b* and *c* the extra amount of stock provided to strengthen the frame for the cross members which are located at these points.

Figure 76 is the rear cross member.

Figures 76*a* and *b* are the front and rear spring hangers respectively; they are steel castings, heat treated, and are riveted to the frame by $\frac{3}{8}$-in. rivets.

Figure 76*c* shows the rear bracket of the front spring.

Figure 76*d* is the rear motor support, a steel casting.

Figure 76*e* gives the details of the body brackets.

Figure 77 is the gusset of the second cross member.

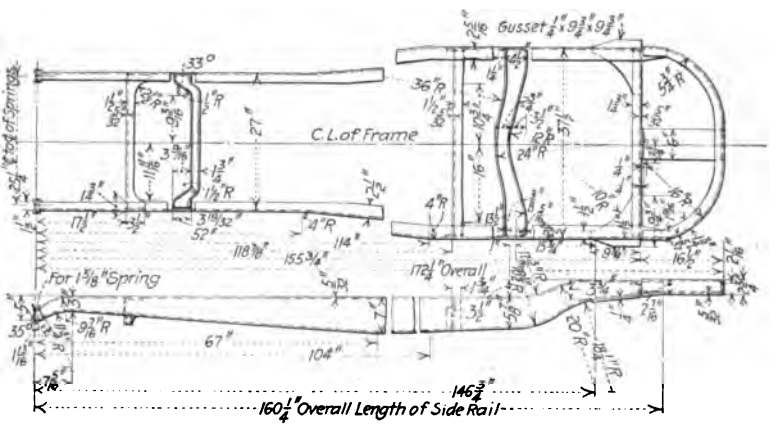


FIG. 78.—Apperson car frame, made by Detroit Pressed Steel Co.

Figure 77*a* is the torque arm bracket located on the second cross member.

Figure 77*b* is the rear socket of the front fender; the front socket being identical, with the exception of dimension *A* which is only $\frac{3}{4}$ in. instead of $1\frac{1}{4}$ in.

Figure 77*c* is one of the muffler supports.

Figure 77*d* is the toe board bracket.

Figure 77*e* is the details of one of the step brackets.

It will be noted from the assembly (Fig. 75), that there is no cross member to support the rear of the motor, but brackets *S* are rivetted to the frame for this purpose. In this case the motor crankcase casting is relied upon to impart rigidity to the two side members. Note in the rear of the frame the double curve,

or "kick-up." The first "kick-up" is to clear the axle, while the frame proper is carried to the end instead of employing gooseneck supports as is frequently done.

Figure 78 is a drawing of the frame of the Apperson car. This frame is manufactured by the Detroit Pressed Steel Co.

FRAME STRESSES

(See Pages 45 to 50 for Stresses in Beams, Etc.)

The stress in the metal of channel frames varies greatly in passenger cars and trucks produced by different makers, and also in the various models of the same manufacturer. Trucks of the same capacity, having vastly different wheel bases, often are found to have channels of the same size. It is true that sometimes a better grade of steel is used by the maker in his trucks of larger capacity, when the section is kept the same. Obviously, if the stresses in the frame are not too high for, let us say, a 7½-ton model, the frame of a 3½-ton truck, having a channel of the same cross section, is certainly too heavy. It can be argued however, that the manufacturer has certain dies for forming the channels and it is less expensive to manufacture a 3½-ton truck with a heavier frame than to have new dies made for a lighter frame. This may be true, but it means more dead weight for the same carrying capacity. What is true of the frame is true of other parts of the machine, and the amount of excessive weight often is considerable, implying a waste of material and constant waste of fuel which might be saved.

An example will now be worked out showing how to calculate frame stresses and for this purpose we will consider a 2-ton truck with a wheelbase of 148 in., and the loads located as shown on Fig. 79. The 4,000-lb. load and the body weight of 960 lb. are distributed over a frame length of 124 in.; one half of this weight, 2,480 lb., is resting on each side member. In practice, the load is not always evenly distributed, but it may be so considered for our purpose, but the factor of safety should be sufficiently high to compensate for uneven load distribution. The driver's seat is 10 in. in front of the body and the gasoline tank is located underneath it. Assuming two persons and the gasoline to weigh 400 lb. we have one-half this load, 200 lb. on each side member. The unit power plant with radiator and water is assumed to weigh 1,000 lb., 60 per cent

of which rests on the rear supports and 40 per cent on the front (the radiator being situated above the front motor support)

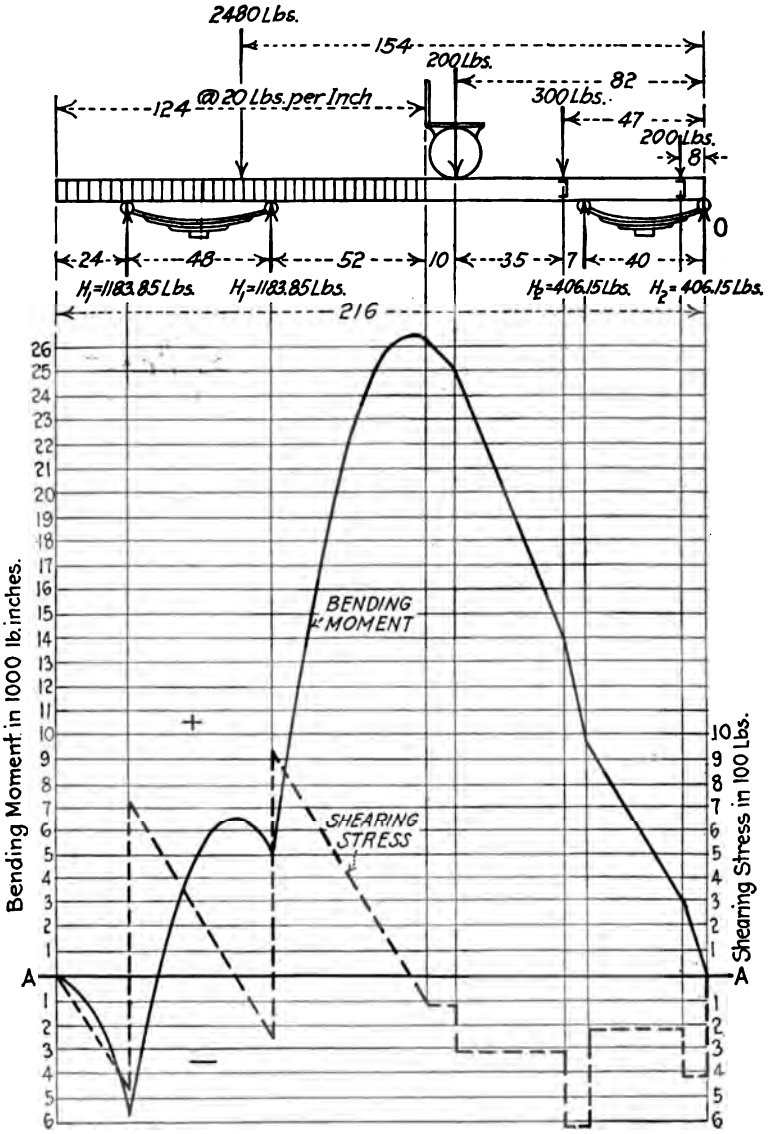


FIG. 79—Illustrating frame stresses.

hence 300 lb. and 200 lb. respectively are resting on each side member at the points indicated in the figure. We did not include

the load due to the torque of the rear axle and when applying the brakes and locking the wheels, which will be referred to later. Thus we have a total load of 3,180 lb. on each side member, supported by two semi-elliptic springs.

The first step is to find the reaction at each spring eye.[1] The distributed load and the body weight are considered for this purpose as concentrated at their centre of gravity, which is 62 in. from the rear end of the frame, and weigh, as was stated 2,480 lb. (on each side member).

Denoting the reactions that are exerted against the frame through their spring eyes, by R_1 and R_2 respectively, the load on each side member is $R_1 + R_2 = 3{,}180$ lb. We are considering R_1 to be located between the two eyes of the rear spring and to be equal $H_1 + H_1$. Thus R_1 is located 168 in. from the front end O. R_2 is determined in like manner for the front spring.

Taking moments about the point O, at the front, by the principle of moments we have $168\ R_1 + 20\ R_2 = (154 \times 2{,}480) + (82 \times 200) + (47 \times 300) + (8 \times 200) = 414{,}100$ lb.-inches. Thus we have two simultaneous equations, $R_1 + R_2 = 3{,}180$, and $168\ R_1 + 20\ R_2 = 414{,}100$. Solving for R_1 in the first equation, we have $R_1 = 3{,}180 - R_2$, and substituting this for R_1 in the second equation we write,

$$168\ (3{,}180 - R_2) + 20\ R_2 = 414{,}100$$

then,

$$534{,}240 - 148\ R_2 = 414{,}100$$

from which $R_2 = 812.3$ lb. In a similar manner $R_1 = 2{,}367.7$ lb., or R_2 can be deducted from the total weight to find R_1.

Since in this example two semi-elliptic springs support each side member, with the axles located centrally between the spring eyes, the reaction at each spring eye is taken as one half that found above.

We therefore have at each rear spring eye $H_1\left(=\frac{R_1}{2}\right)$, 1,183.85 lb. and 406.15 lb. $\left(\frac{R_2}{2}\right)$ at each front spring eye, as noted on the drawing.

The next procedure is to determine the bending moments at various points along the frame; to make the diagram fairly accurate, the bending moments are determined at each 6 in.

[1] See also, C. F. CLEAVER, *Automobile Engineer*, August, 1912, for an analysis of frame stresses.

along the entire length. The 2-ton load and the body weight are considered as being evenly distributed over the 124-in. length, and are $\frac{2,480}{124} = 20$ lb. per inch. In practice, the frame weight per inch of length may be included to ensure greater accuracy, also other concentrated weights that may be supported by the frame. It was stated before that the bending moment B at any vertical section of a beam (at any point along the beam) equals the algebraic sum of all the moments, (*i.e.*, the negative moments must be deducted from the positive moments) of the external forces about that section, acting either to the right or to the left of the section.

The following table gives the values so calculated, but the bending moments are worked out for a few points to make the method clear.

TABLE VI

At (inches)	B = (lb-inchs)	At (inches)	B = (lb-inches)	At (inches)	B = (lb-inches)
6	− 360	78	10,191	150	20,066.0
12	−1,440	84	14,677	156	18,192.0
18	−3,240	90	18,443	162	16,318.0
24	−5,760	96	21,490	168	14,444.0
30	−1,897	102	23,816	174	11,071.0
36	1,246	108	25,422	180	9,021.0
42	3,669	114	26,308	186	7,785.0
48	5,372	120	26,474	192	6.548.0
54	6,356	124	26,185	198	5,310.7
60	6,619	132	25,287	204	4,073.8
66	6,162	138	23,813	210	2,436.9
72	4,985	144	21,940	216	0.0

The moments which would tend to produce clockwise rotation at the left of the section are positive, and those tending to produce counter-clockwise rotation, negative. The reverse of this is true when the moments on the right of the section are considered.

Beginning from the left and up to the first reaction H_1, the moments can be calculated from the formula for uniformly loaded cantilever beams, $B = \frac{1}{2} LW$, where L is the length of the cantilever beam up to the point at which the bending moment B is to be determined, and W is the total weight on the cantilever beam up to that point. Twenty-four inches from the left (over the first reaction H_1) we have $B = -\frac{1}{2} LW = -\frac{1}{2} \times 24 \times (24 \times 20) = -5,760$ lb.-inches. (The load per inch is 20 lb.)

Beyond the 24-in. point, to the right, the moments due to the reactions must be considered.

At 42 in. from the left we have in addition to the negative moment, the positive moment due to the reaction at the spring eye, thus $B = -(\frac{1}{2}) \times 42 \times (42 \times 20) + 18H_1 = 3{,}669$ lb.-inches.

At 84 in., $M = -(\frac{1}{2}) \times 84 \times (84 \times 20) + 60H_1 + 12H_1 = 14{,}677$ lb.-inches.

In this manner all the figures in the table were obtained.

From these values the bending moment diagram (Fig. 79) is drawn as shown by the full line curve. The vertical or ordinate values are laid off to some convenient scale. In the drawing each space represents 1,000 lb.-inches. The bending moment, at every point of the frame, is evident from a glance at this curve. The ordinates above the base line *A-A* represent the positive moments, and those below, the negative values.

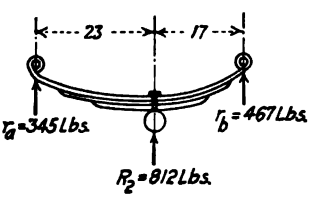


FIG. 80.—Reaction on offset spring eyes.

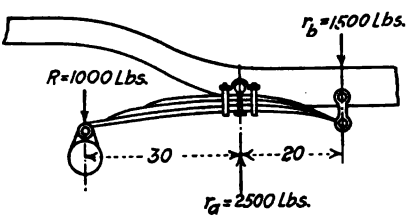


FIG. 81.—Reactions on cantilever spring.

When the axle is not at the centre of the spring, that is, with offset springs, such as are used frequently with front suspension, moments must be calculated to ascertain the reactions between the spring eyes and the frame. Let us suppose, for example, the front spring eccentrated as shown in (Fig. 80), where the axle is 17 in. to the rear of the front eye. By taking moments about r_a we have, $40r_b = 23R_2$, from which $r_b = \dfrac{23 \times 812}{40} = 467$ lb., and $r_a = 345$ lb. When calculating the stresses in a frame the reactions must be established carefully to obtain accurate results.

With half-elliptic *cantilever* springs the reactions at the frame supports may be considerably higher than the axle reaction. For instance, taking the example shown in (Fig. 81) where this type of spring is attached to the frame at a point 30 in. from the axle and the other end is anchored to the frame 20 in. from the spring

bolt. Assuming the axle reaction R to be 1,000 lb., moments are taken about the front end r_b to find the reaction at r_a.

Thus, $50 \times 1{,}000 = 20r_a$, hence $r_a = \frac{50{,}000}{20} = 2{,}500$ lb.; r_b acts downward like an extra load on the frame and equals $\frac{30 \times 2500}{50} = 1{,}500$ lb. Thus it is evident that the type of spring employed has an important bearing on frame stresses.

Shearing Forces.—In a frame channel only the vertical shearing stresses need be considered. The magnitude of the shearing force at any vertical section of the frame is the algebraic sum of the shearing forces acting on either side of that section. To make this clear the shearing stresses at one point are worked out to show how the values were obtained from which the shearing force diagram (shown in dotted lines) was drawn.

At the front spring eye of the rear spring, 72 in. from the rear of the frame, the total load tending to cause a negative shear of the section is $72 \times 20 = -1{,}440$ lb. Just at the left of the section we have the positive shear of 1,183.85 lb. (the first reaction), and $-1{,}440 + 1{,}183.85 = -256.15$ lb.; just at the right of this section the shear is $-1{,}440 + 1{,}183.85 + 1{,}183.85 = 927.7$ lb., hence, at this section the shearing stress changes from negative to positive. Attention is drawn to the fact that at the maximum bending moments the shear is always zero.

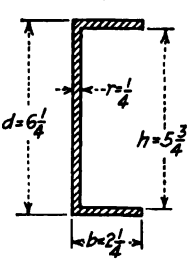


FIG. 82.—Frame channel section.

The stresses in the frame metal can be determined from the bending moment diagram, when the dimensions of the channel are known. The stress in any layer of a beam varies directly as the distance from the neutral axis, the maximum fiber stress being of course in the upper and lower flanges of the channel. Knowing the moment of resistance, or the bending moment B at any section of the channel, the maximum stress can be determined by the well known formula $B = SZ$, hence $S = \frac{B}{Z}$ where S is the maximum fiber stress (per square inch) in the section, and Z is the section modulus.

Let us suppose that the channel used in this frame has the section and dimensions shown in (Fig. 82), *i.e.*, it is 6¼ in. high, 2¼ in. wide, and is made of ¼-in. stock. The section modulus

of a channel is (see table III page 49) $Z = \frac{bd^3 - h^3(b-t).}{6d}$ Hence, for the channel section above, we have

$$Z = \frac{2\frac{1}{4} \times 6\frac{1}{4}^3 - 5\frac{3}{4}^3 \times (2\frac{1}{4} - \frac{1}{4})}{6 \times 6\frac{1}{4}} = 4.50$$

The maximum bending moment is at a point about 120 in. from the left end of the frame, as shown on the diagram; here B = 26,474 lb.-inches. The maximum stress $S = \frac{B}{Z} = \frac{26,474}{4.50} =$ 5,900 lb. approximately. However, it must be remembered that this stress of 5,900 lb. in the outer fiber is induced when the load is static on the frame, as shown. Under running conditions the springs will sometimes be deflected to about twice their static deflection, due to impact, or as much as the clearance will allow, and since the deflection in a spring is proportional to the load, the maximum load on the frame must be assumed to be double the normal (static) load shown. Hence, the bending moment, and consequently the maximum stress in the frame are double the amount found above. The stress is considerably more when the frame hits the axle, as it does sometimes, and the load on the frame arising from the torque arm or torque tube, when applying the brakes, may alter the stress distribution in the side members very materially, as is shown under torque arms, Chapt. IX. Taking the stress at twice the amount calculated gives a maximum stress of 11,800 lb. per square inch, and if the ultimate strength of the material is, for instance, 60,000 lb. per square inch, the factor of safety is $\frac{60,000}{11,800} = 5.1$ approximately. This is none too much, when the overloads to which a truck is subjected are considered, and that the load on the frame due to the torque, when the truck is being stopped, with the wheels locked, was not taken into account.

By studying the bending moment diagram, one can see at a glance where the frame can be lightened without sacrificing strength. The shearing stress is greatest near the spring shackles; the maximum being less than 1,000 lb. in the example given. The ultimate shearing strength of this quality of steel is about 50,000 lb. per square inch, hence the section in our case is ample; there is more danger from crippling of the web, then shearing of the metal.

CHAPTER IV

CLUTCHES

In motor vehicles, driven by internal combustion engines, a clutch must be provided for disconnecting the engine from the transmission, at will, to slow down or stop the car without "shifting" gears. With sliding gear transmissions, some means must also be provided for gradually and smoothly establishing connection between the engine and the transmission, and through the latter with the road wheels; if this is not done the car will start forward with a jerk and the shifting of gears is liable to be a difficult operation. A friction clutch is admirably suited to accomplish these objects. In this clutch, two or more surfaces are held in frictional contact with each other through the medium of one or more springs. Connection between these frictional surfaces must not be established too suddenly, (the clutch must slip a trifle before taking permanent hold) else it will cause damage in time to the motor as well as to the entire transmission mechanism and the tires. If the clutch grips jerkily or harshly it will cause slippage between tires and ground, and the life of the tires will thereby be reduced quite materially. The clutch must also be so designed that it may be disengaged completely, rapidly, and without dragging, to permit shifting gears quickly and noiselessly.

There are various types of clutches on the market, the most popular being the multiple dry disk and the dry plate (single disk) clutch; the cone clutch is also used in some models in this country (it is more popular abroad) likewise the multiple disk clutch running in oil. According to the 1922 statistical number of *Automotive Industries*, over 94 per cent of the passenger car models and 98 per cent of the truck models use the disk clutch, either the multiple or single disk.

The amount of power which a clutch can transmit depends, on the pressure between the frictional surfaces, on the coefficient of friction, and on the average distance or mean radius of the frictional surfaces from the axis.

If the spring pressures exerted in a horizontal direction, (see

Fig. 83), or in line with the shaft, is P; if A is the angle of the cone, which is also the angle of the direction of the force F (from a vertical line) pressing the clutch cone against the clutch surface, and if F is the total normal (actual) pressure between the clutch surfaces, then $\sin A = \frac{P}{F}$; $F = \frac{P}{\sin A}$ and $P = F \sin A$. The actual resistance by which the frictional surfaces are prevented from slipping on each other, is equal to the coefficient of friction c, multiplied by the force F (the pressure between the surfaces), thus Fc and this equals, $\frac{Pc}{\sin A}$ since $F = \frac{P}{\sin A}$. If the mean radius of the cone is r, and T the torque which the clutch is

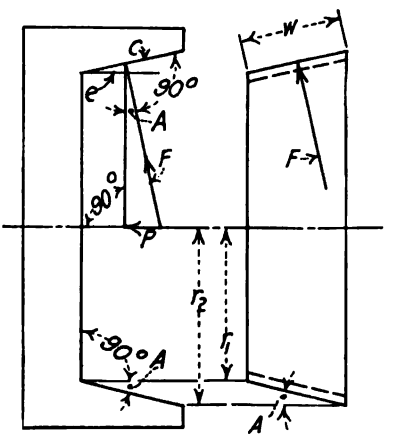


FIG. 83.—Cone clutch.

capable of transmitting, $T = Fcr$ pound inches (F being expressed in pounds and r in inches). For practical purposes it is sufficiently accurate to take $r = \frac{r_1 + r_2}{2}$.

It was shown in Vol. I, Engines, Chap. XX, that the torque T of an engine in pound-feet $T = \frac{\text{hp.} \times 5252.1}{R}$ hence T in pound-inches $\frac{\text{hp.} \times 5252.1 \times 12}{R} = \frac{\text{hp.} \times 63{,}025}{R}$, where R is the number of revolutions per minute. Knowing the break horse power (hp.) of an engine at various speeds, it is possible to find the maximum torque (this is not at the maximum hp.) and to design a clutch which will transmit such maximum torque. The coeffi-

cient of friction between leather and cast iron or between raybestos or asbestos fabrics and metal, may betaken as about .25. The next step in the design of a clutch is to determine the area of the frictional surfaces. The larger the area, the smalleer is the necessary pressure per square inch to transmit a certain power, but the larger the area, the heavier is the clutch cone. A heavier clutch cone renders the shifting of gears less easy for there is a heavier rotating mass whose speed has to be changed, therefore, the inertia of the clutch cone should not be greater than necessary. (This will be discussed later in this chapter.) The pressure per square inch of clutch surface area in practice, is as a rule, between 4 lb. and 10 lb. The higher this pressure the sooner will the clutch surface wear out. In cone clutches the pressure is higher than in multiple disk or single plate clutches, since in the latter a larger friction surface is readily obtainable.

In a cone clutch, the clutch surface area $a = 2\pi rw$, w being the width of the clutch surface and r the mean radius of the cone. If f is the normal pressure (the actual pressure between the friction surfaces) per square inch of friction surface, the total pressure $F = fa$, and therefore $T = facr$, and substituting the value for a as above, $T = f2\pi\, rwcr = f2\pi r^2wc$. From this formula anyone of the unknown quantities can be found if the others are known. For example, the maximum torque of a certain motor is 1,600 lb.-inches, the radius r is largely determined from the size of the flywheel, and let us assume it is 5 in.,(which is rather smaller than the average), find w. f may be taken at 8 lb. per square inch, and the coefficient of friction e at .25. By simple algebra,

$$w = \frac{T}{f2\pi r^2c} = \frac{1{,}600}{8 \times 2 \times 3.1416 \times 25 \times .25} = 5.1 \text{ in.}$$

Such a width is too great for an ordinary clutch, but it can only be reduced if the radius is increased (in practice such a clutch would have a larger radius) or by increasing the pressure f. Increasing the pressure will cause the clutch facing to wear more rapidly, but to obviate a very wide clutch a higher pressure is often used. In practice, ordinarily, a multiplie disk or plate clutch is employed for such a motor if r can not be increased. It was shown before that the spring pressure required $P = F \sin A$; this is the pressure necessary to hold the clutch in engagement at a certain engine torque, but a higher pressure is required when the clutch surfaces are first brought into contact, for then there is slippage between

the clutch surfaces. The coefficient of friction of rest is greater than the coefficient of friction of motion, hence when the surfaces are slipping, the frictional force between them is considerably lower. For this reason spring pressure greater than P is required, when the two parts of the clutch are first brought into engagement, and until both parts revolve at the same speed.

Sometimes it is necessary to find the maximum hp., which a clutch is capable of transmitting at a certain speed. We saw before that the torque

$$T = \frac{\text{hp.} \times 63{,}025}{R} \text{ lb.-inches,}$$

from which

$$\text{hp.} = \frac{TR}{63{,}025}.$$

By substituting the value of T found before,

$$\text{hp.} = \frac{FcrR}{63{,}025},$$

from which

$$F = \frac{\text{hp.} \times 63{,}025}{crR}.$$

Since $P = F \sin A$, we can write

$$P = \frac{\text{hp.} \times 63{,}025 \times \textit{sine}\, A}{crR}.$$

(See also page106.)

Coefficient of Friction. Take a stationary plate with surface s, (Fig. 84), and place upon it a block b which is forced against the plate by the normal pressure F (that is, at right angles to s) by means of a weight. If a force E is exerted against the block at right angles to F, evidently the friction between b and s will tend to prevent b from being moved, but by slowly increasing the magnitude of E, a point will be reached when b will begin to move, after which a somewhat lesser force E suffices to move it slowly at a uniform speed.

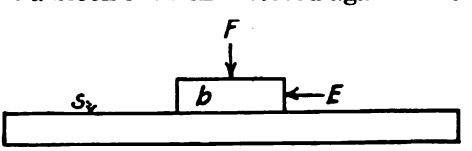


FIG. 84.—Illustrating coefficient of friction.

The ratio between the force E, which slowly moves the body b at a uniform speed, and the normal pressure (the force F acting at right angles to s, which forces the surfaces together) is called

the coefficient of friction (of motion) c and this is usually employed. Hence $c = \frac{E}{F}$, and $E = cF$. For instance, if the block b is pressed against s with a pressure or force F of 100 lb., and if the force E required to move the block b horizontally at a slow uniform speed, is 25 lb., the coefficient of friction $c = \frac{25}{100} = .25$. The coefficient of friction varies with different materials. The average value for asbestos friction materials on steel, as used in disk clutches, is about .35, while when the friction material is partly worn, and when the surfaces are not perfectly dry, when oil finds its way to the friction surfaces, the coefficient will be about .25. It is, therefore, recommended that the value of .25 be used in calculations incidental to the design of clutches.

The coefficient of friction for metal to metal is very low, therefore, a very much higher normal pressure per square inch is required, else very large friction surfaces are necessary. It might be stated that the friction between two surfaces is independent of their area, *viz.*, if the area of b (Fig. 84) were made smaller, or larger, so long as the total normal pressure F remains the same, it will require the same force E to set it into motion. Therefore, in a clutch, whether the area of the friction facing is large or small, it will require the same total normal pressure F, but if the area is small, the pressure per square inch will be larger and the friction facing will wear out more rapidly. The friction varies directly with the normal pressure, hence the larger the value of F the greater will be E. The condition of the surfaces of the material also affect the friction; it is evident that if the surfaces are rough, E will be greater than if they were smooth, and the coefficient of friction will be higher. However the size of the area does not affect the force E. Assume, for instance, that b had a surface of 10 sq. in. and the normal pressure F were 100 lb.; in such case, we would have $\frac{100}{10}$ or a pressure f of 10 lb. per square inch of surface, but $F = f \times \text{area} = 100$. If the area were now reduced to 5, we would have only one-half the number of square inches but the pressure per square inch would be $\frac{100}{5}$ or 20 lb. that is, double the amount, but, pressure per square inch $\times$ area = 100, thus, the total friction (or Force E) remains the same. (This is not always true,

for under special conditions, the coefficient of friction will increase with an increase in pressure, or a decrease in area.)

Angle of Cone.—For cast iron on leather or asbestos facing, the angle of the cone varies between 10° to 15,° 12° being about the average. For metal to metal surfaces a smaller angle is used, as a rule from 9° to 11°.

The advantage of a smaller angle is that a smaller spring pressure is required (it was shown before, the spring pressure $P = F \sin A$, and if A is smaller P is smaller) but a smaller angle is more liable to cause the clutch to drag or stick, when thrown out; it has a reduced clearance for a given axial movement of the cone, and it is apt to grip more suddenly.

Spring Force Required.—As was stated before, the normal pressure per square inch varies ordinarily between 5 and 12 lb. in leather or asbestos faced clutches, while in metal to metal clutches running in oil, it is about 35 to 40 lb. smce the coefficient of friction is much smaller, and the clutch would slip under a comparatively small torque unless the pressure were considerably higher than that used in leather or asbestos faced clutches. In order to find the total spring pressure when the maximum pressure f per square inch has been determined, we may proceed as follows: In the foregoing example we had, $r = 5$ in., $w = 5.1$ in. and $f = 8$ lb.

$F = f\,a = 8 \times 2\,\pi\,r\,w = 1{,}280$ lb. Suppose the angle of the cone to be $12\frac{1}{2}$,° the sine of $12\frac{1}{2}° = .2164$. Hence in our example the necessary spring pressure $P = F$ sine $12\frac{1}{2}° = 1{,}280 \times .2164 = 277$ lb. Instead of a single spring, a plurality may be employed. In practice a higher pressure is necessary when the friction surfaces are first brought together and springs are as a rule, made adjustable to vary the pressure, as required. A difference in the material of the clutch facing, will make a difference in the coefficient of friction, likewise a facing which has become polished by use will be different from one which is still rough. On the other hand, if the spring pressure is too great the clutch will grip too suddenly and this will impart a jerky movement to the car and a severe strain on the joints, gears, etc.

Examples of Clutches.—In most of the motor vehicles produced in the United States "dry" multiple disk clutches are provided. The friction surfaces are woven asbestos fabric or molded asbestos composition.

The plate or single disk clutch requires a higher pressure per square inch between the friction surfaces than the multiple disk type to transmit the power without slipping. It will therefore wear out more rapidly, everything else being equal, than when a larger number of disks is used with a smaller pressure per square inch. If the diameter of the disk is increased, the pressure *f* can be, of course, reduced.

Figure 85 shows the clutch-case assembly of the Northway pressed steel clutch, which was used on the 1920 eight-cylinder Oldsmobile.

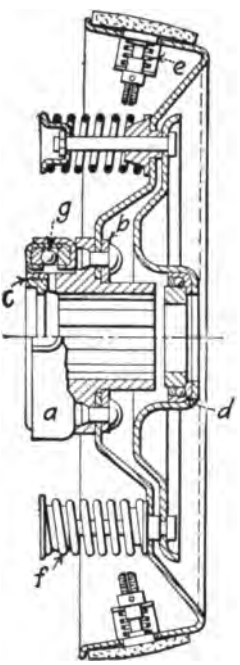


FIG. 85.—Northway pressed steel clutch.

Figure 86 is a detail of the cone, a steel stamping, made of No. 14 (.078) U.S.S. gauge stock. Figure 86*a*, the clutch cone hub made of No. 1114 S.A.E. steel (screw stock) 4⅛ in. in diameter. It gives all the details and the tolerances where accuracy is required.

Figure 86*b* is the pressed steel clutch spring support. Figure 86*c*, the clutch release bearing retainer marked *a* on the cone assembly. Figure 86*d* is a detail of the clutch cone backing ring shown at *b*. Figure 86*e* is the clutch release bearing nut, shown at *c*, which is made of seamless steel tubing.

Figure 86*f*, the clutch spring support ball bearing, located at *d*. Figure 86*g*, the clutch cone leather, which is pressed on the cone and then finished; underneath the leather, at six places, there are spring controlled plungers *e* for raising the leather at such places when the clutch is disengaged and to make gradual engagement when the clutch is first thrown in. Instead of leather, asbestos fabric is often employed, as it can withstand higher temperatures.

Figures 86*h* and 86*i* are the cup and bolt guides for the clutch spring *f* of which four are employed to hold the clutch in engagement. Around the clutch hub the ball bearing is located, shown at *g*; by means of an actuating fork or lever, controlled by the clutch pedal, the clutch release bearing retainer is forced away from the clutch, and this causes the clutch springs to be com-

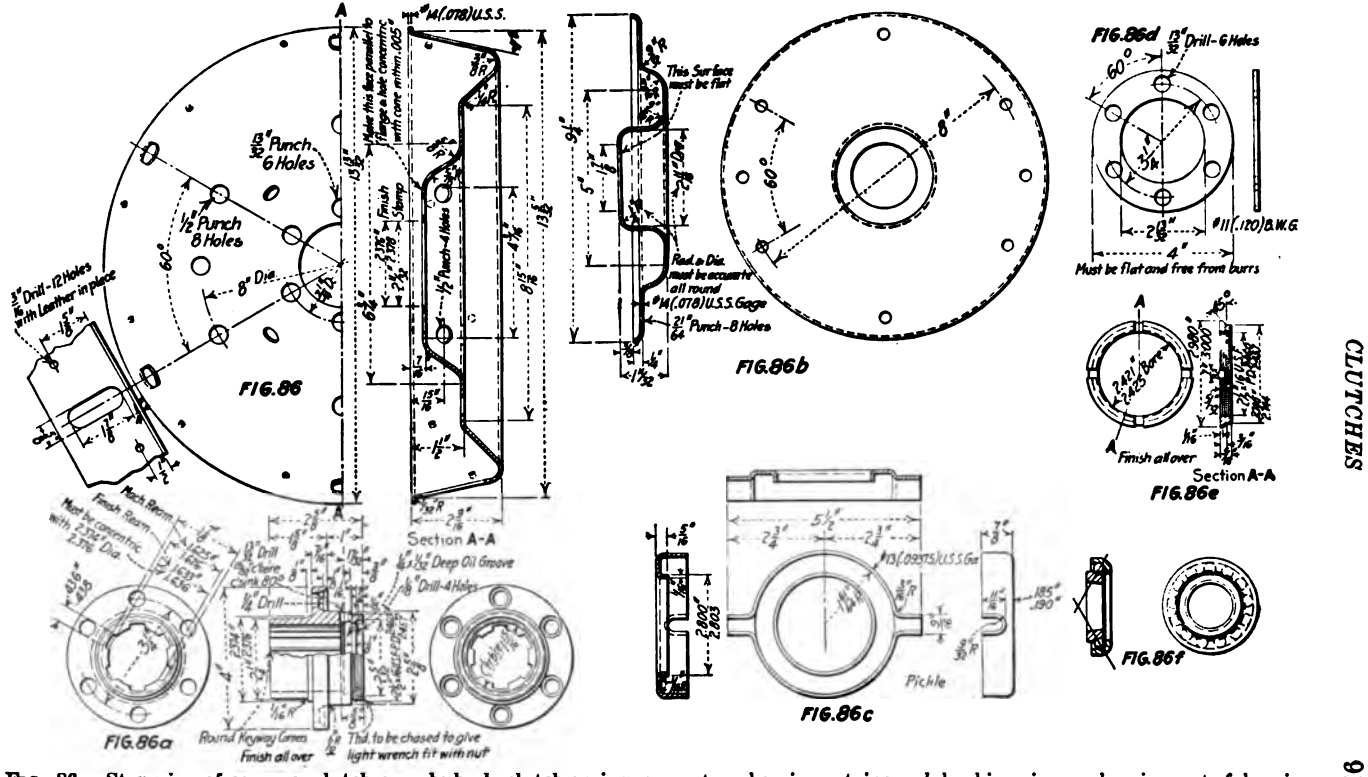


FIG. 86.—Stamping of cone; *a*, clutch cone hub; *b*, clutch spring support; *c*, bearing retainer; *d*, backing ring; *e*, bearing nut; *f*, bearing (Northway clutch).

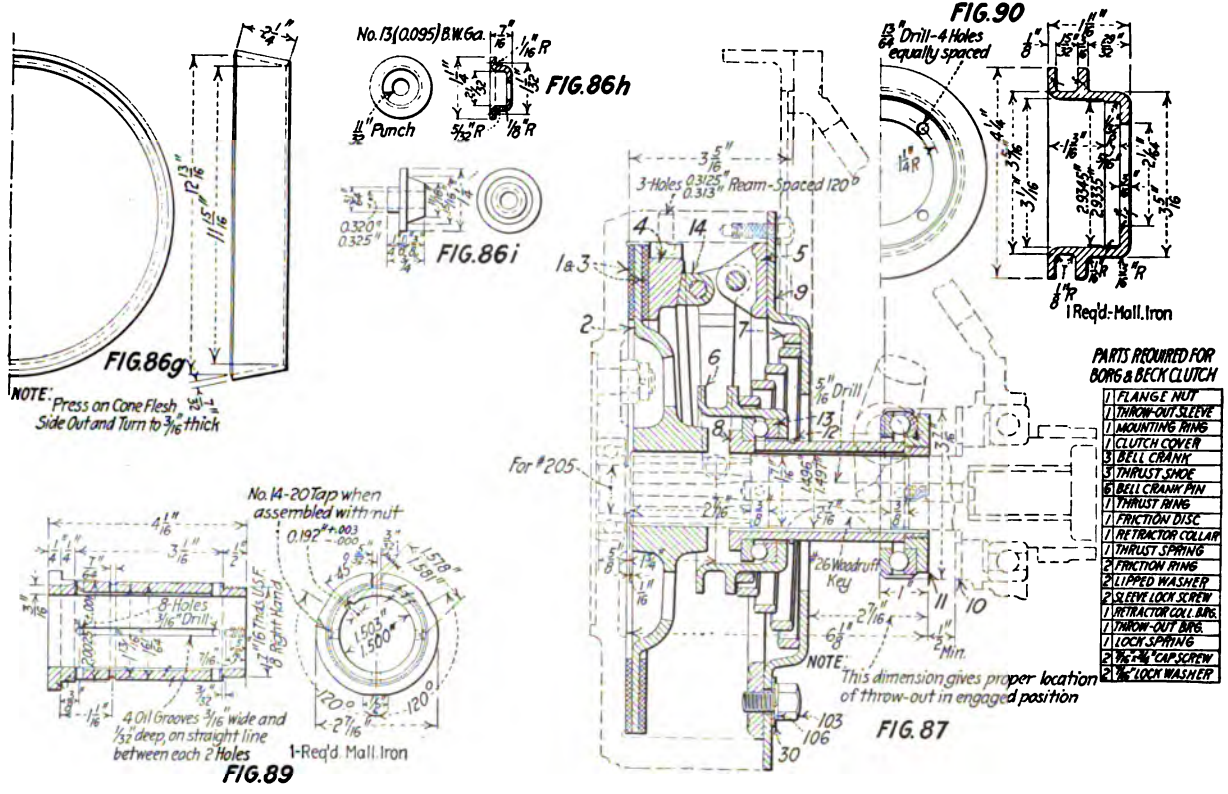
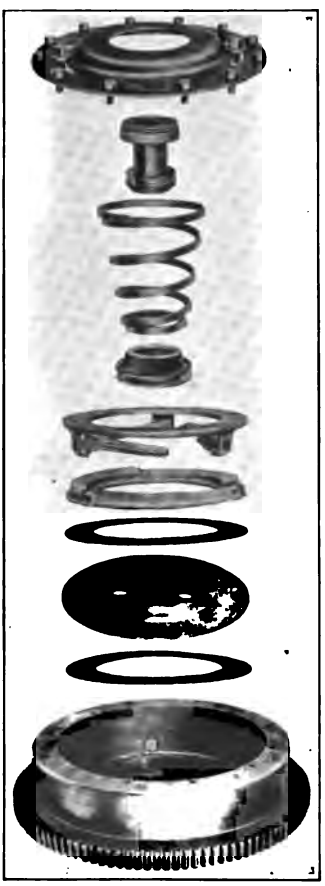


FIG. 86*g*.—Clutch cone leather; *h*, cup; *i*, bolt guide (Northway clutch). FIG. 87.—Borg and Beck single plate clutch. FIG. 89.—Sleeve. FIG. 90.—Collar.

pressed thereby releasing the grip between the clutch cone and the flywheel.

In some clutches cork inserts are used in connection with the leather, to obtain gradual engagement. The leather is fastened to the cone by means of rivets, countersunk, so that the surface



9. Clutch cover—it bolts to the flywheel—note adjustment bolts in slots.

8. Sleeve—it carries retractor collar—note thrust bearings.

7. Spring—is compressed between cover and retractor collar—it operates bell crank levers.

6. Retractor collar—ends of bell cranks engage in slot.

5. Mounting ring—fastens to cover with adjustment bolts— it carries the bell crank levers.

4. Thrust ring—note inclined face toward center and adjustment inclines.

3. Friction ring—copper reinforced woven asbestos.

2. Friction disc — or dry-plate— keyed to shaft by 10 splines.

1. Friction ring—a duplicate of No. 3. These rings take the wear.

A. Flywheel ready for clutch—note dowel pins that carry thrust ring (4).

FIG. 88.

be below that of the leather; sometimes *T* bolts are used for holding the leather down.

Disk Clutches. (See also "Clutches" in chapter on Transmission).—A more recent tendency in design is the dry disk clutch, in which one or more disks are rotating and engage other disks lined with woven or moulded asbestos friction fabrics. The

latest friction material to make its appearance on the market is the endless moulded asbestos clutch facing for which greater durability and greater accuracy as to dimensions is claimed and this renders possible closer tolerances in clutch design.

Dry disk clutches are divided into two groups, single plate and multiple disk.

The most widely used example of the former is the Borg & Beck clutch, (Fig. 87), in which a single dry-plate or disk is locked to, and rotates with, the flywheel by the gradual application of pressure between two asbestos rings, thereby driving the clutch shaft that is connected to the transmission.

The pressure is brought to bear on the drive disk by the positive lever action of three arms or bell cranks, which virtually act as wedges, being forced into action by the thrust of a powerful clutch spring. Figure 88 shows the clutch disassembled; the flywheel is made into a hollow drum which contains the parts between it and the cover *9*.

The friction disk *2* and the asbestos rings, *1* and *3*, the clutch shaft and its sleeve *8*, are the only parts not anchored to the flywheel and float in the drum until they are locked with it by the wedge action of the bell cranks carried on the mounting ring *5*.

When the clutch pedal is depressed, the sleeve *8* (Fig. 89) and collar *6* (Fig. 90) are pulled back, moving the bell cranks and consequently releasing the wedge pressure on the thrust ring *4* (Fig. 91); when released, the spring thrust moves the bell cranks forward which exert the pressure that locks the drive disk.

It is to be noted that the thrust ring *4* is an inclined plane, tapered toward the inner edge. The thrust shoes *14* (Fig. 91*a*) attached to the bell cranks by means of pins (Fig 91*b*), mount this inclined surface as the spring thrust is exerted and gradually increase the pressure on the friction rings, *1* and *3*. This causes the friction disk to begin to move; at first slowly and then with increasing speed until the pressure of the thrust ring finally locks it to the flywheel.

An annular ball bearing mounted on the sleeve *8* inside of the retractor collar *6* allows the shaft and sleeve to come to rest with the friction disc when the clutch is thrown out. A similar ball thrust bearing is mounted on the opposite end of the sleeve and takes the thrust of the clutch spring as the sleeve is pulled backward under the pressure of the yoke of the clutch pedal.

Oil for the sleeve and bearings is provided from the transmission, it flows through the hollow clutch shaft and through holes bored in the sleeve itself.

Upon the thrust ring tremendous power is exerted through the toggle action of three shoes mounted on the long lever arms which are actuated by the coiled spring, (Fig. 91*c*).

The friction or main driving disk *2* has a ten-spline broached hub which rides on the clutch shaft and thus rotates with it.

The clutch is engaged when the foot lever is released and the coiled spring exerts its thrust upon the throw-out sleeve. The

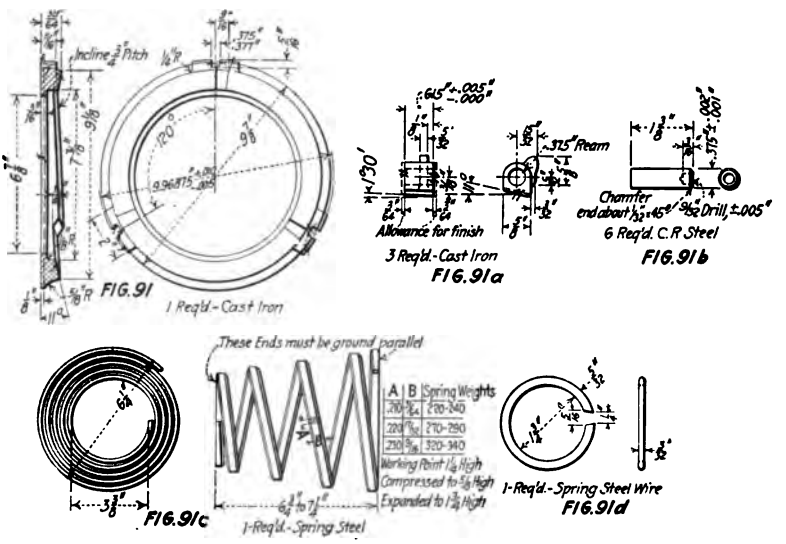


FIG. 91.—Thrust ring; *a*, thrust shoe; *b*, shoe pin; *c*, clutch spring; *d*, bearing lock spring (Borg and Beck clutch).

sleeve carries the grooved retractor collar in which the lower ends of the bell crank thrust levers are set. The outer ends of the bell crank thrust levers are in the form of Y-shaped trunnions, one end of which is bolted to an adjustable mounting ring and the other carrying the swivel-mounted thrust shoes.

Thus, a powerful wedge action is obtained which is brought into action by the thrust of the clutch spring. When the foot lever is applied, the throw-out sleeve and throw-out collar are retracted, telescoping the spiraled coils of the spring into almost a single plane. The lever movement of the bell cranks causes the thrust shoes to recede toward the inner periphery of the thrust ring

which releases the pressure on the friction disk and allows it to come to rest. The friction disk, being light in weight, is easily stopped by the brake action of the sleeve *8*, with flange nut *11*, rubbing against the housing hub brake surface *10*.

One face of the thrust ring *4*, by which it presses against friction disk *3*, is a plane surface, the outer face being a double inclined plane, (see Fig. 91), inclined radially to take the wedge action of the thrust shoes, and circumferentially to allow for adjustment.

When it becomes desirable to compensate for working wear, the adjustment is made by unloosening the two adjustment bolts which project through the cover plate *9* and which carry the adjusting or mounting ring. Turning this adjusting ring in a clockwise direction changes the relation of the thrust shoes to the thrust ring so that the distance in which the wedge action takes place is shortened and thus the grip on the friction surfaces is increased. Figure 91*d*, (*12* on the assembly, Fig. 87) is the inner bearing lock spring, to keep bearing *13* in place when the clutch is disassembled.

The clutch release bearing offers more or less difficulty to proper lubrication on account of its inaccessability. A number of concerns use the method here indicated, or else drill through the main drive gear shaft of the transmission (see chapter on Transmission) and then drill cross holes leading to the bearings. While some manufacturers claim they obtain proper lubrication in this manner, others claim this method inefficient in that either too much oil is fed through, which will eventually drain the transmission case, or no oil will flow through at all, especially in cold weather. Sometimes special means are employed to force the oil into the drilled shank of the main gear drive shaft, but in such case there may be danger of too much flow of oil.

In the clutch manufactured by the Warner Gear Co. (see Fig. 147, assembly of transmission and clutch) a stationary bearing retainer (Fig. 92) is used, which is held against rotation by being supported on the inner ends of the clutch throw-out shafts, which project through the inner faces of the clutch release yoke (Fig. 92*a*); the lugs *L* of the retainer resting on the shaft ends. This retainer is provided with an oil pocket, as shown, in upper view through which the oil, flows to a reservoir at the bottom. The short throw-out shaft (Fig. 92*b*) is drilled through the center and tapped for an oil connection, from which an oil

lead is carried to one of the rear motor legs or to any other convenient place.

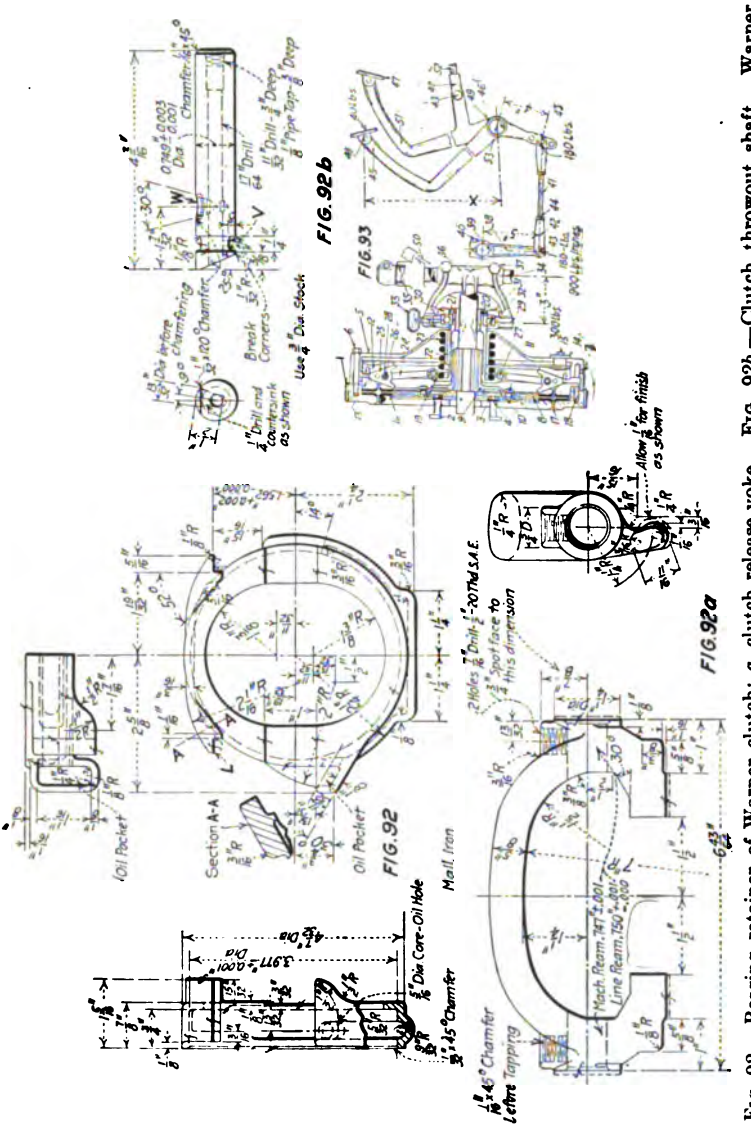


FIG. 92.—Bearing retainer of Warner clutch; *a*, clutch release yoke. FIG. 92*b*.—Clutch throwout shaft. Warner clutch. FIG. 93.—Clutch of White 3½- and 5-ton truck.

By occasionally pouring a little oil into this lead, it runs, through the drilled throw-out shaft, drips off at the groove

marked *V*, through the oil pocket of the retainer into the small reservoir, whence the oil finds its way into the bearing. *W* shows a tapered hole for a set screw (Fig. 149) whereby the shafts (the one shown, and the longer one through which the clutch release yoke is operated) are fastened to the yoke.

Another type of single plate clutch where a toggle wedge is employed to apply pressure between the friction plate shoes and the single steel disk is shown in (Fig. 93), which is used on the White 5-ton truck. This clutch is running in oil, and is not a dry-plate clutch like the Borg & Beck.

The action of this clutch is evident from the figure. When the clutch foot lever *45* is depressed, lever *38*, which is keyed and clamped to shaft *34*, moves the clutch pull-out shoes *29*, away from the clutch, and this in turn exerts a pull on the clutch toggle spider *19* through ball bearing *32*. As the toggle spider *19* slides toward the right it withdraws the wedges *25* from the clutch plate rings *12* and *13*, in this manner releasing the pressure between the disk *8* and friction plate shoes *16*.

Ref No.	Number Per Car	Name
1	1	Clutch flywheel.
2	6	Clutch flywheel bolts.
3	6	Clutch flywheel bolt nuts.
4	6	Clutch flywheel bolt nut lock washers.
5	1	Clutch flywheel cover.
6	8	Clutch flywheel cover bolts.
	8	Clutch flywheel cover bolt washers.
7	2	Clutch flywheel plugs.
8	1	Clutch friction plate.
9	1	Clutch friction plate hub.
10	8	Clutch friction plate hub bolts.
11	8	Clutch friction plate hub bolt nuts.
12	1	Clutch friction plate ring (notched).
13	1	Clutch friction plate ring (plain).
14	1	Clutch friction plate ring lock.
15	2	Clutch friction plate ring lock screws.
	2	Clutch friction plate ring lock screw lock washers.
16	2	Clutch friction plate shoes.
17	18	Clutch friction plate shoe rivets ($\frac{3}{16} \times 1$ in.).
18	8	Clutch friction plate springs.
19	1	Clutch toggle spider.
20	1	Clutch toggle spider lock nut.
21	1	Clutch toggle spider lock nut lock wire.
22	1	Clutch toggle spider key.
23	1	Clutch toggle spider spring.
24	12	Clutch toggle spider straps.
25	6	Clutch toggle spider strap wedges.
26	6	Clutch toggle spider strap pins (short).
27	6	Clutch toggle spider strap pins (long).
28	36	Clutch toggle spider strap pin washers.
29	1	Clutch pull-out shoe.
30	1	Clutch pull-out shoe cover.
31	4	Clutch pull-out shoe cover screws.
	4	Clutch pull-out shoe cover screw lock washers.
32	1	Clutch pull-out shoe bearing.
33	1	Clutch pull-out shoe oil cup.
34	1	Clutch pull-out shoe shaft.
35	2	Clutch pull-out shoe shaft bushings.
36	1	Clutch pull-out shoe shaft yoke.
37	2	Clutch pull-out shoe shaft yoke screws.
	2	Clutch pull-out shoe shaft yoke screw nuts.
	2	Clutch pull-out shoe shaft yoke screw nut lock washers.

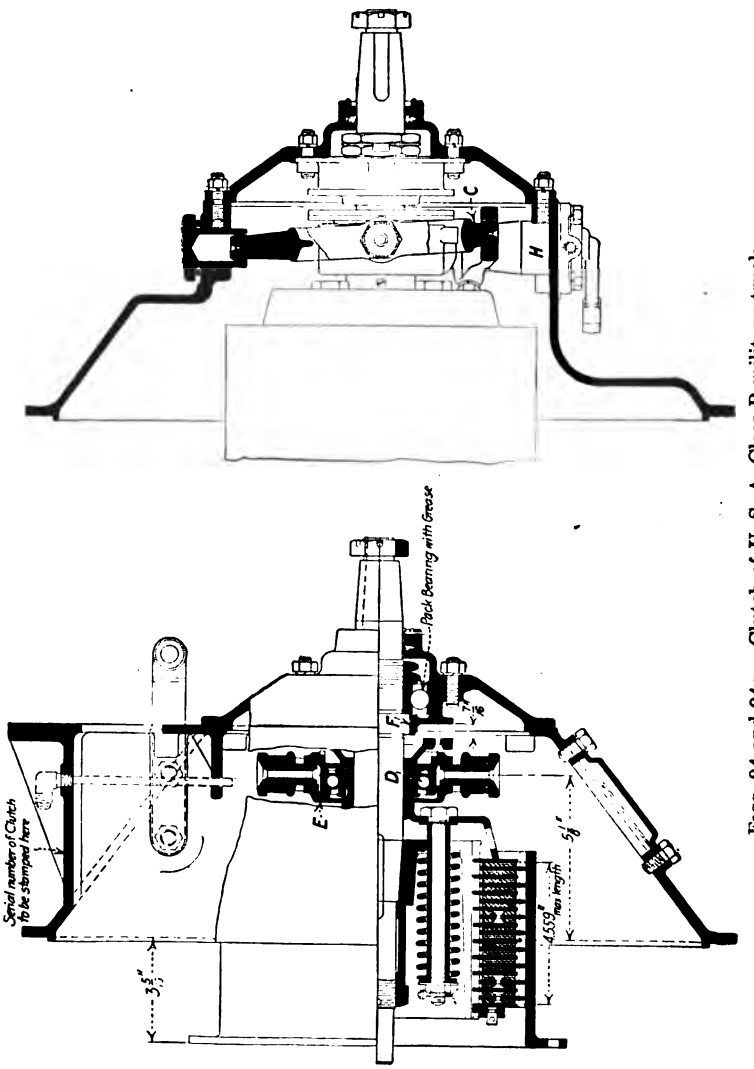
Ref No.	Number Per Car	Name
38	1	Clutch pull-out shaft lever.
39	1	Clutch pull-out shaft lever key.
40	1	Clutch pull-out shaft lever binder bolt.
	1	Clutch pull-out shaft lever binder bolt lock washer.
41	1	Clutch pull-out shaft lever connection.
42	2	Clutch pull-out shaft lever rod ends.
43	2	Clutch pull-out shaft lever rod end pins.
	2	Clutch pull-out shaft lever rod end pin nuts.
44	2	Clutch pull-out shaft lever rod end jam nuts.
45	1	Clutch foot lever.
	1	Clutch foot lever adjusting stop.
	1	Clutch foot lever adjusting screw.
	1	Clutch foot lever adjusting screw jam nut.
	2	Clutch foot lever bracket screws.
46	2	Clutch foot lever bushings.
47	2	Clutch foot lever pads.
48	2	Clutch foot lever pad rivets.
49	2	Clutch foot lever oilers.
50	1	Clutch oil tube.
51	1	Brake foot lever.
52	1	Brake foot lever rod ($1\frac{7}{32} \times 64\frac{3}{4}$ in.).

Figures 94, 94*a* and 95, give assembly views of the clutch of the U. S. A. Class B military truck. This, as mentioned before, is nominally a 3½-ton truck, but is equivalent to a 5-ton commercial model.

In this clutch there are nine driving disks faced with asbestos friction rings, at each side, to which they are riveted, (see Fig. 95*a*). The rivet head is $\frac{3}{64}$ in. below the friction surface. The entire thickness on the driving disk is indicated. Figures 95*b* and 95*c* are details of the driving and driven disks respectively. In this clutch there are 18 ring surfaces in contact with the 10 driven disks. (The first and the last driven disks contact the friction material on only one of their sides, see Fig. 94.) The driving disks are provided with 88 teeth at their outer periphery, which engage with similar teeth of the clutch driving drum, (Fig. 96). This drum is attached to the flywheel by eight $\frac{3}{8}$-in. bolts.

The 10 driven disks have 60 teeth at their inner circle which mesh with similar teeth of the clutch driven drum, (Fig. 95*d*). All the teeth are cut (not stamped) for accuracy; if the teeth are too tight, the clutch drags, and if too loose, it rattles or knocks. The drum is keyed to the clutch shaft (Fig. 97) and is drawn up tight by a taper fit and nut as can be seen from the assembly drawing. The disks are normally held in engagement by three coil springs held in place by spacers, (Fig. 96*a*), bolted to the clutch pressure plate (Fig. 96*b*). The latter is attached to the hub and brake flange, (Fig. 96*c*). This flange is provided with a friction facing, forming the clutch brake, which comes into action when the clutch is released entirely. The clutch is actuated through the clutch release bracket shaft, (Fig. 96*d*), (*A*, in Fig.

95) to which lever *B* is keyed which engages *C* of the release yoke, (Fig. 98), and this acts on the ball bearing *D*. The latter engages the hub 96*c*, and the pressure plate 96*b* which compresses the



FIGS. 94 and 94*a*.—Clutch of U. S. A. Class B military truck.

springs through the spring bolts. Figure 98*a* is the aluminum end plate of the crankcase and clutch housing. Figure 98*b* is the release bearing retainer (marked *E* in Fig. 94), (Fig. 98*c*), the

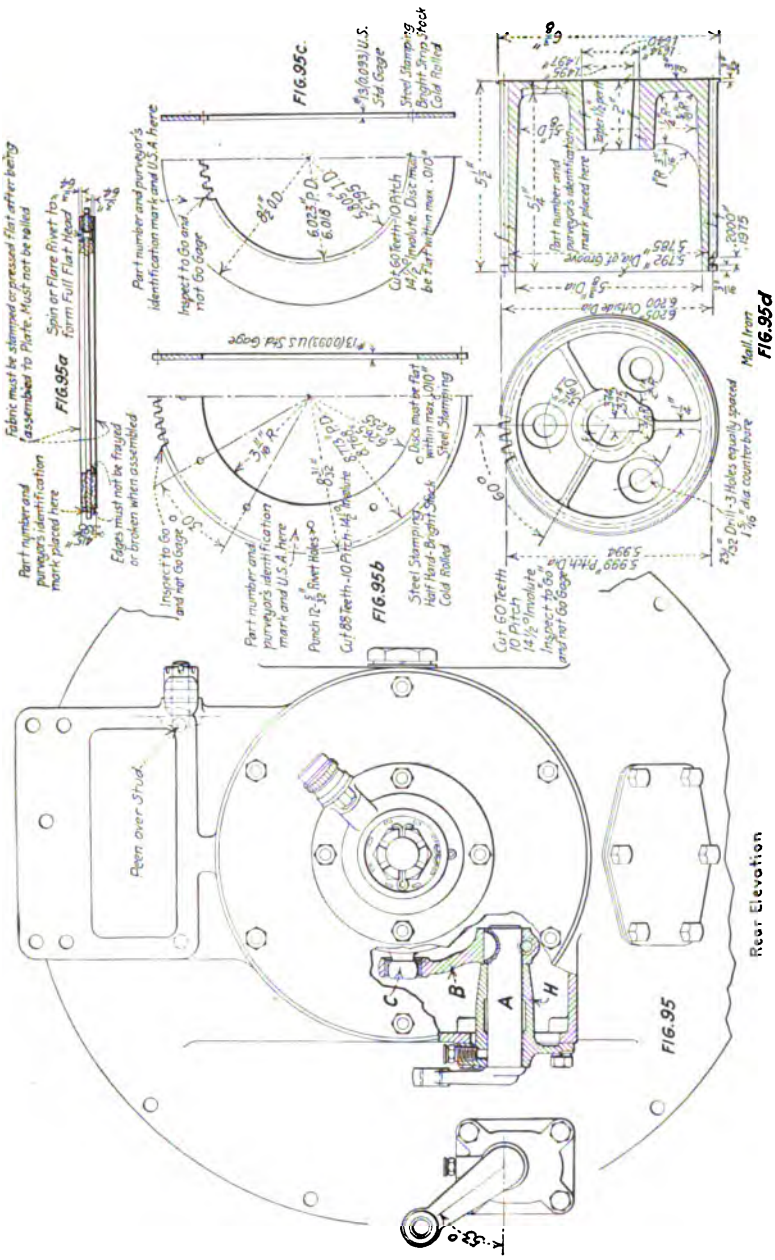

Fabric must be stamped or pressed Flat after being assembled to Plate. Must not be rolled
Spin or Flare Rivet to form Full Flat Head
Part number and purveyor's identification mark placed here
Edges must not be frayed or broken when assembled
F16.95a
Inspect to Go and not Go Gage
Part number and purveyor's identification mark and U.S.A. here
Punch 12-5/32" Rivet Holes
Cut 88 Teeth-10 Pitch-14½° Involute
F16.95b
Steel Stamping Half Hard-Bright Stock Cold Rolled
Discs must be flat within max. .010" Steel Stamping
13(0.093) U.S. Std Gage
F16.95c
Steel Stamping Bright Strip Stock Cold Rolled
Part number and purveyor's identification mark placed here
Cut 60 Teeth-10 Pitch 14½° Involute Disc must be Flat within max. .010"
Cut 60 Teeth 10 Pitch 14½° Involute
Inspect to Go and not Go Gage
Drill-3 Holes equally spaced 1⅝" dia counterbore
Mail from
F16.95d
Peen over Stud
Rear Elevation
F16.95
A
B
C
H

FIG. 95.—Class B clutch; *a*, asbestos friction ring; *b* and *c*, driving and driven disks; *d*, clutch driven drum.

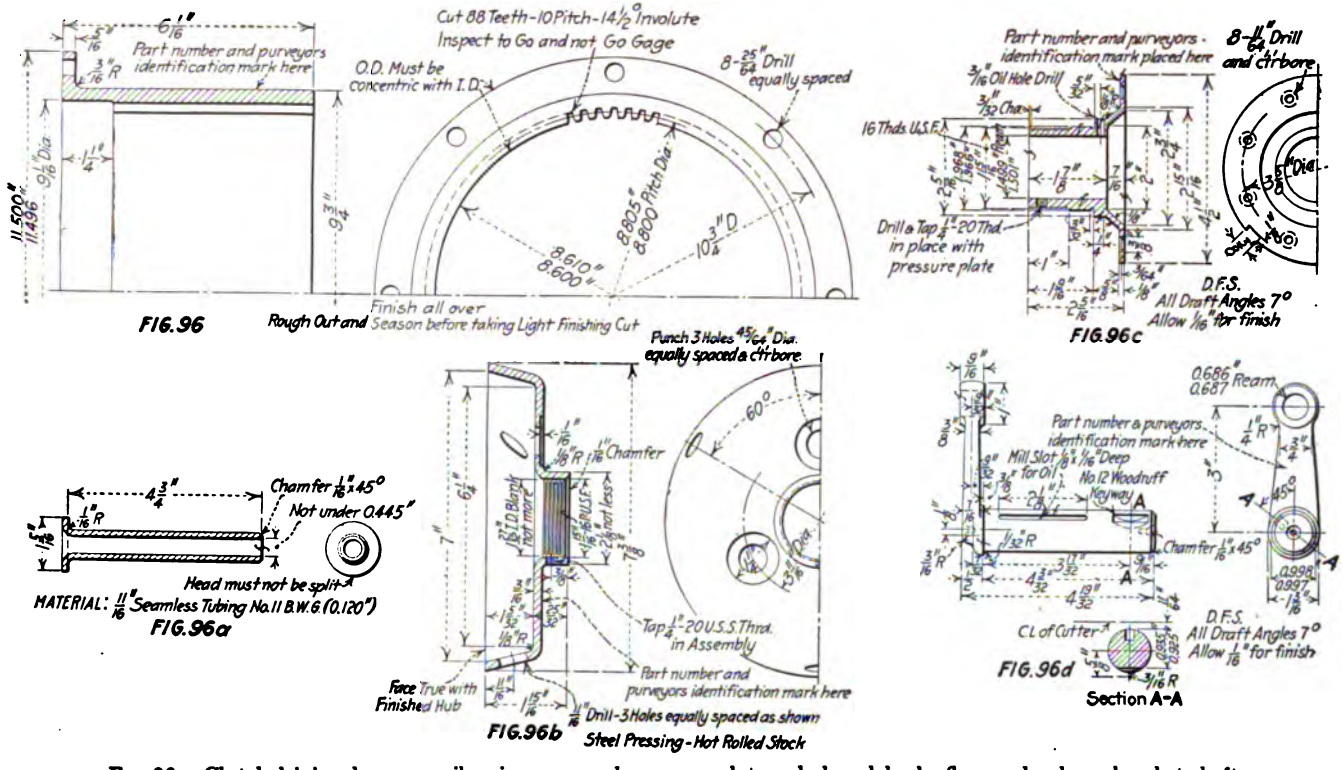

Part number and purveyors identification mark here
Cut 88 Teeth - 10 Pitch - 14½° Involute
Inspect to Go and not Go Gage
O.D. Must be concentric with I.D.
8-25/64" Drill equally spaced
8.805" 8.800" Pitch Dia.
8.610" 8.600"
10 3/4" D
Finish all over
Rough Out and Season before taking Light Finishing Cut
FIG. 96
Part number and purveyors identification mark placed here
8-11/64" Drill and c'bore
3/16" Oil Hole Drill
3/32" Chs.
16 Thds. U.S.F.
Drill & Tap 1/4"-20 Thd. in place with pressure plate
D.F.S.
All Draft Angles 7°
Allow 1/16" for finish
FIG. 96c
Chamfer 1/16" x 45°
Not under 0.445"
Head must not be split
MATERIAL: 11/16" Seamless Tubing No. 11 B.W.G. (0.120")
FIG. 96a
Punch 3 Holes 45/64" Dia. equally spaced a c'bore
1/16" Chamfer
Tap 1/4"-20 U.S.S. Thrd. in Assembly
Part number and purveyors identification mark here
Face True with Finished Hub
11/16" Drill-3 Holes equally spaced as shown
Steel Pressing - Hot Rolled Stock
FIG. 96b
0.686" 0.687" Ream
Part number & purveyors identification mark here
Mill Slot 1/8" x 1/16" Deep for Oil
No. 12 Woodruff Keyway
Chamfer 1/16" x 45°
0.998" 0.997"
CL of Cutter
D.F.S.
All Draft Angles 7°
Allow 1/16" for finish
FIG. 96d
Section A-A

FIG. 96.—Clutch driving drum; *a*, coil spring spacers; *b*, pressure plate; *c*, hub and brake flange; *d*, release bracket shaft.

rear bearing retainer F, against whose surface the clutch brake acts. Figure 98*d* is the clutch release rocker shaft bracket, marked H on the assemblies.

In multiple disk clutches, the friction area depends on the number of fractional surfaces, and the mean circumference. If r_1 is the inner, and r_2 the outer radius, then the mean radius r of the disk, is $\frac{r_1 + r_2}{2}$, and the area $a = 2\pi rwn$, w being the width and equals $r_2 - r_1$, while n represents the number of surfaces in frictional contact.

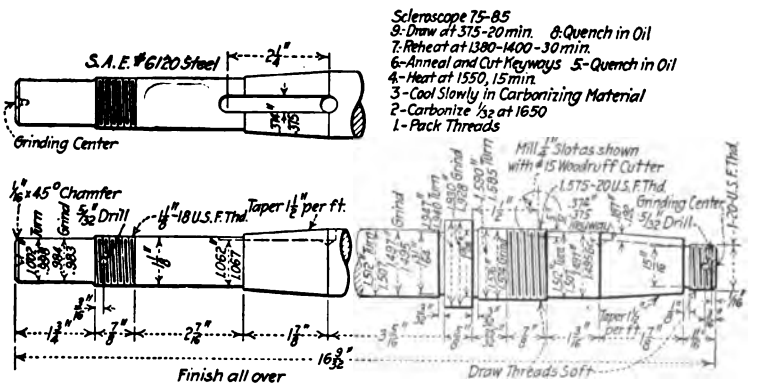


FIG. 97.—Clutch shaft, Class B truck.

As in the case of the cone clutch, the torque T which the clutch can transmit $= facr$, and substituting the value for the area found above we have $T = f\,2\pi r^2 wcn$.

From the hp. formula of a motor we have hp. $= \frac{TR}{63{,}025}$, hence $T = \frac{63{,}025 \text{ hp.}}{R}$, which is the same as that found for the cone clutch. It should be noted however that a in the clutch formula for disk clutches includes the number n of frictional surfaces, and that F is the same as P (see Fig. 83) since the spring pressure and the normal pressure are the same, as the spring acts at right angles to the friction surfaces.

As an example take the U. S. A. Class B military truck motor having a maximum torque of 2,800 lb.-inches; the multiple disk clutch used has 18 friction surfaces of a disk width w of 1 in., and a mean friction disk radius of 3.75 in.

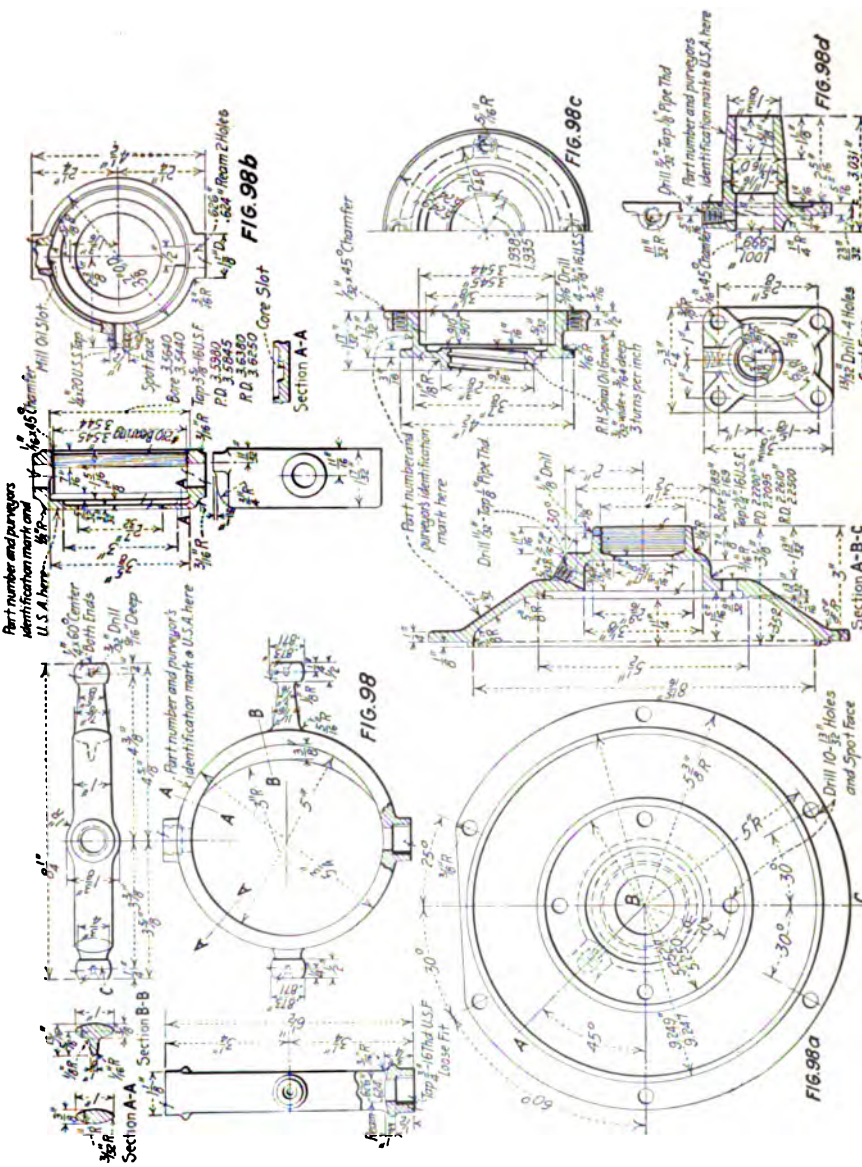


FIG. 98.—Release yoke; *a*, end plate (aluminum); *b*, release bearing retainer; *c*, rear bearing retainer; *d*, release shaft bracket.

The area a of the entire friction surface equals $2\pi rwn = 2 \times 3.14159 \times 3.75 \times 1 \times 18 = 425$ sq. in.

The engine torque $T = 2{,}800 = facr = f \times 425 \times .25 \times 3.75$,

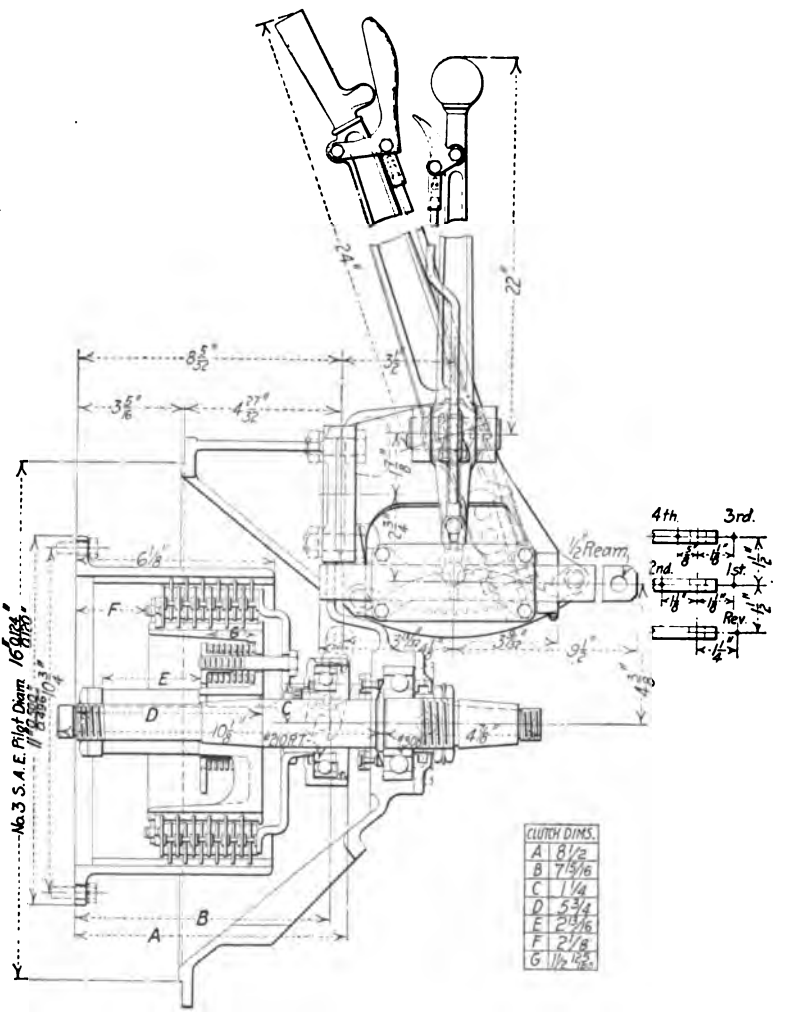


FIG. 99.—Detlaff clutch.

hence $f = \dfrac{2{,}800}{425 \times .25 \times 3.75} = 7$ lb. per square inch of surface.

In a multiple disk clutch F is not the pressure per square inch, multiplied by the total friction area (as in the cone clutch) but

$F = f \times$ area of one disk; thus the spring pressure. $F = 7 \times 23.6 = 165$ lb. approximately.

(In a disk clutch if the pressure is exerted at the pressure plate the pressure is transmitted equally to all the surfaces.)

Ordinarily, the friction material is attached to the driving disks (which rotate with the flywheel) and not to the driven disks, for the lighter the latter the sooner will they stop spinning when the clutch is thrown out, since their inertia is less.

Figure 99 is the Detlaff clutch which is similar to that of the Class B truck. Figure 100 is a photograph of the clutch driven drum *H* which is keyed to the clutch shaft (not shown in this

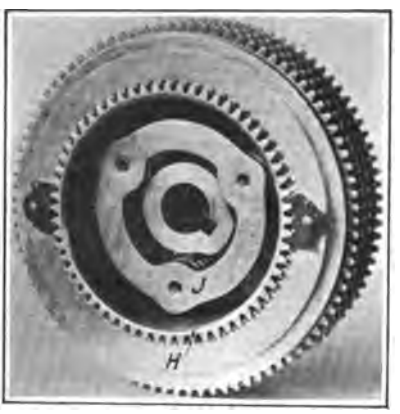


FIG. 100.—Clutch driven drum (Detlaff).

picture); on this drum the disks can be seen assembled, also the three springs and the spring plate *J*. Figure 99, discloses also the bell-housing (No. 3, S.A.E.) and control set with the shifting control of the transmission gears; (see also Chap. VI). The capacity of the clutch depends on the number of disks; it is increased or decreased by an addition or reduction in the number of driving disks. It is stated by the makers that the five-disk clutch is suitable for a motor torque of 3,000 lb.-inches approximately.

Figure 101 is the clutch of the ¾- to 1½-ton truck manufactured by the International Harvester Co. Special attention is called to the clearances indicated for the various parts, the oilless bearing on which the clutch drum rotates when the clutch is thrown out, and the location of the clutch throw-out shaft *b* and the clutch brake *c*. Figure 102 is the model G.C.L. clutch, manufactured by Fuller & Sons Mfg. Co.; it is suitable for a motor

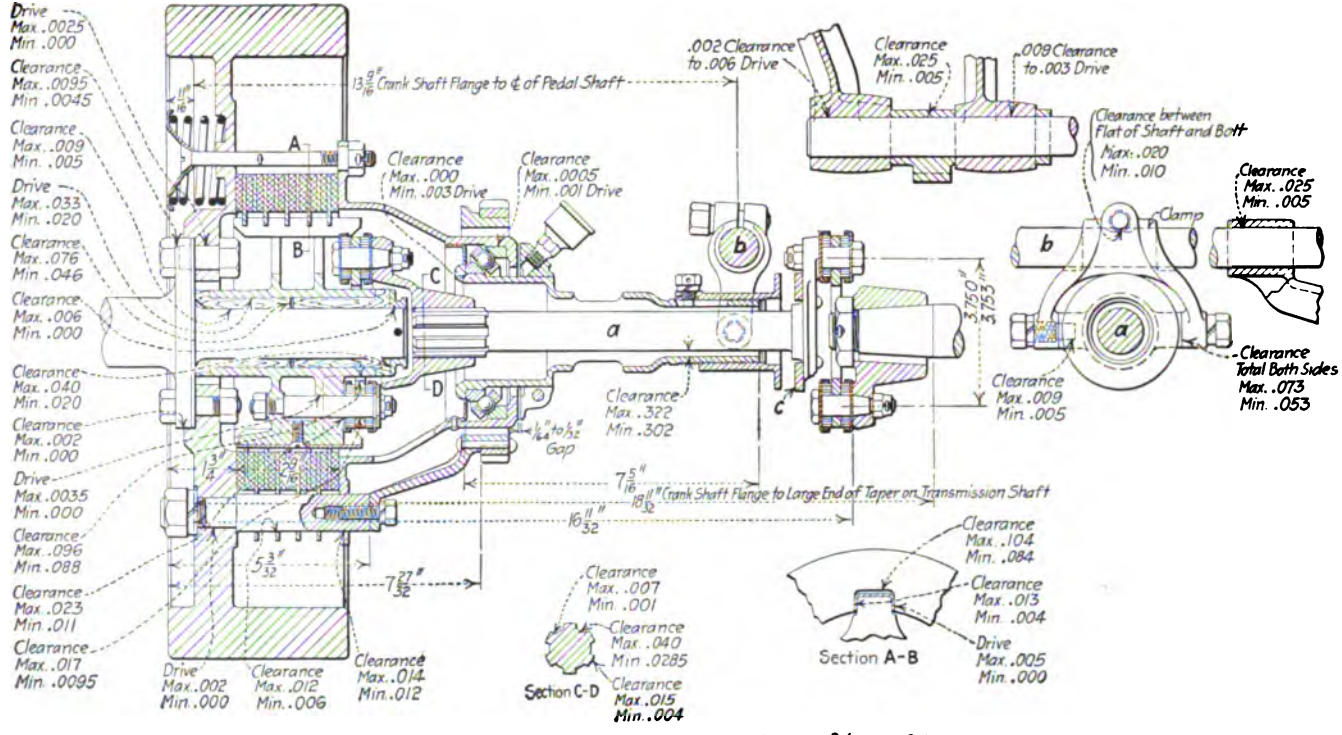
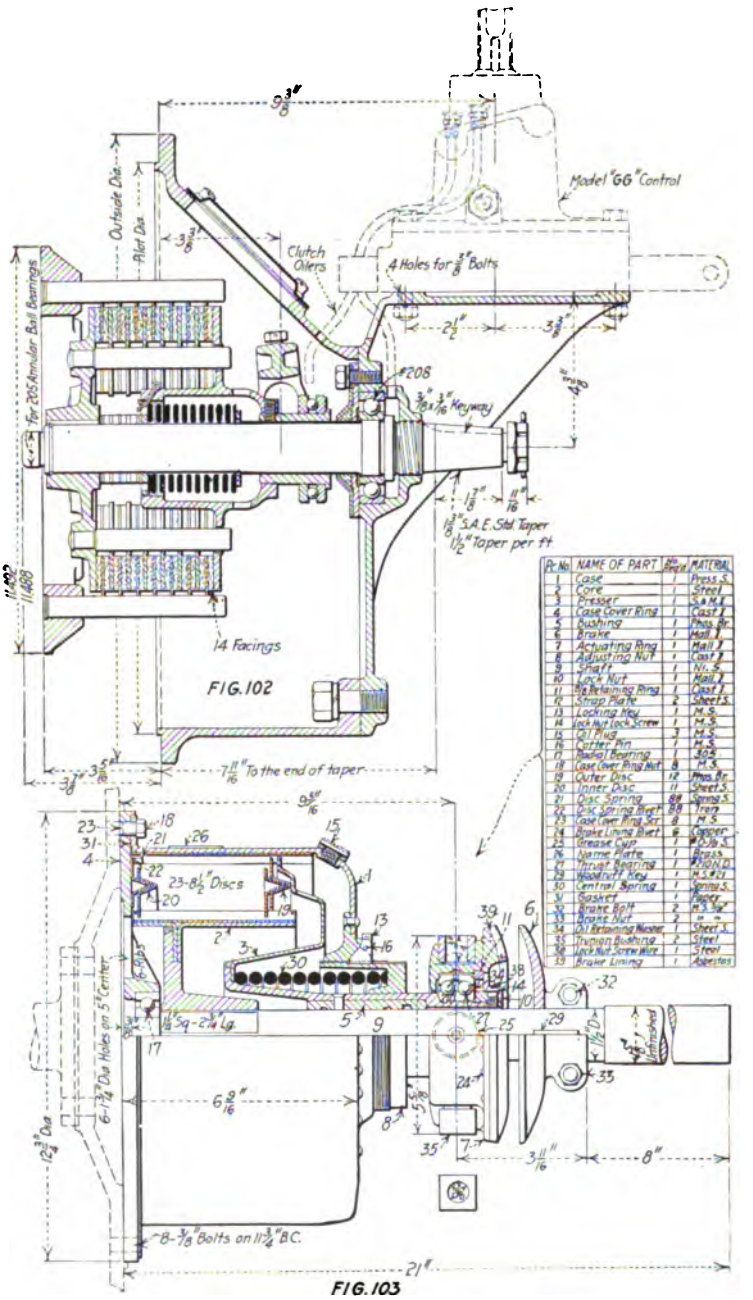

Drive Max. .0025 Min. .000
Clearance Max. .0095 Min. .0045
Clearance Max. .009 Min. .005
Drive Max. .033 Min. .020
Clearance Max. .076 Min. .046
Clearance Max. .006 Min. .000
Clearance Max. .040 Min. .020
Clearance Max. .002 Min. .000
Drive Max. .0035 Min. .000
Clearance Max. .096 Min. .088
Clearance Max. .023 Min. .011
Clearance Max. .017 Min. .0095
13 9/16" Crank Shaft Flange to ℄ of Pedal Shaft
.002 Clearance to .006 Drive
Clearance Max. .025 Min. .005
.009 Clearance to .003 Drive
Clearance between Flat of Shaft and Bolt Max. .020 Min. .010
Clearance Max. .025 Min. .005
Clamp
Clearance Max. .000 Min. .003 Drive
Clearance Max. .0005 Min. .001 Drive
A
B
C
D
a
b
c
3.750"
3.753"
Clearance Max. .009 Min. .005
Clearance Total Both Sides Max. .073 Min. .053
Clearance Max. .322 Min. .302
1/4" to 5/32" Gap
7 5/16"
18 11/32"
Crank Shaft Flange to Large End of Taper on Transmission Shaft
16 11/32"
1 3/4"
5 3/32"
7 27/32"
Drive Max. .002 Min. .000
Clearance Max. .012 Min. .006
Clearance Max. .014 Min. .012
Clearance Max. .007 Min. .001
Clearance Max. .040 Min .0285
Section C-D
Clearance Max. .015 Min. .004
Clearance Max. .104 Min. .084
Clearance Max. .013 Min. .004
Drive Max. .005 Min. .000
Section A-B

FIG. 101.—Clutch of International Harvester Co. for ¾- to 1½-ton truck.



Pc. No	NAME OF PART	No. Reqd	MATERIAL
1	Case	1	Press S.
2	Core	1	Steel
3	Presser	1	S.A.E.
4	Case Cover Ring	1	Cast I.
5	Bushing	1	Phos. Br.
6	Brake	1	Mall. I.
7	Actuating Ring	1	Mall. I.
8	Adjusting Nut	1	Cast I.
9	Shaft	1	Ni. S.
10	Lock Nut	1	Mall. I.
11	Ball Retaining Ring	1	Cast I.
12	Strap Plate	2	Sheet S.
13	Locking Key	1	M.S.
14	Lock Nut Lock Screw	1	M.S.
15	Oil Plug	3	M.S.
16	Cotter Pin	1	M.S.
17	Thrust Bearing	1	305
18	Case Cover Ring Nut	8	M.S.
19	Outer Disc	12	Phos. Br.
20	Inner Disc	11	Sheet S.
21	Disc Spring	88	Spring S.
22	Disc Spring Rivet	88	Iron
23	Case Cover Ring Scr.	8	M.S.
24	Brake Lining Rivet	6	Copper
25	Grease Cup	1	1/8 in. S.
26	Name Plate	1	Brass
27	Thrust Bearing	1	Fiber
28	Headless Key	1	M.S. #3
29	Central Spring	1	Spring S.
30	Gasket	1	Paper
31	Brake Bolt	2	M.S. Sq.
32	Brake Nut	2	" "
33	Oil Retaining Washer	1	Felt
34	Thrust Bushing	2	Steel
35	Lock Nut Screw Wire	1	Steel
36	Brake Lining	1	Asbestos

Fig. 102.—Clutch of Fuller & Sons Mfg. Co. (1½- to 3½-ton truck).
Fig. 103.—Hele-Shaw clutch (for 40 h.p. motor or 3- to 4-ton truck).

up to 350. cu. in. piston displacement, or a 1½- to 3½-ton truck.

Figure 103 is the Hele-Shaw clutch (manufactured by the Merchants & Evans Co.) suitable for a 40 hp. motor, or a 3- to 4-ton truck. Figure 104 is a detail drawing of the clutch core, (Fig. 104*a*) the phosphor bronze driving disk, and (Fig. 104*b*) the inner, or driven, disk made of steel. The Hele-Shaw multiple disk clutch, is made up of phosphor bronze and cold rolled steel "V" grooved disks; the number of disks and their diameters varying with the hp. to be transmitted and the class of service for which it is intended. With V-disks a small spring pressure

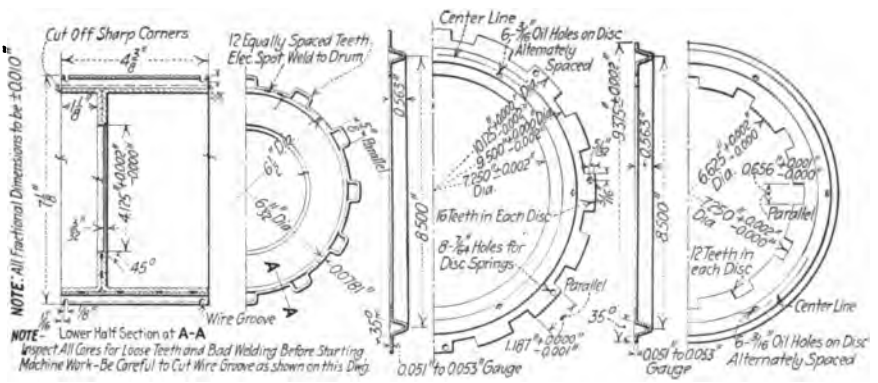

Cut Off Sharp Corners
12 Equally Spaced Teeth
Elec. Spot Weld to Drum
Center Line
6-3/16" Oil Holes on Disc
Alternately Spaced
16 Teeth in Each Disc
8-7/64" Holes for Disc Springs
Parallel
12 Teeth in each Disc
Wire Groove
Lower Half Section at A-A
Inspect All Cores for Loose Teeth and Bad Welding Before Starting Machine Work - Be Careful to Cut Wire Groove as shown on this Dwg.
0.051" to 0.053" Gauge
0.051" to 0.053" Gauge
6-3/16" Oil Holes on Disc Alternately Spaced
NOTE: All fractional Dimensions to be ±0.010"
NOTE

FIG. 104. FIG. 104*a*. FIG. 104*b*.

FIG. 104.—Clutch core; *a*, driving disk (bronze); *b*, driven disk (steel). Hele-Shaw clutch.

exerts a great normal pressure between the plates (see description of cone clutch) and a larger disk area is obtainable with a given disk width. The disks can also be made thinner as the V-shape renders them stronger and prevents distortion.

With multiple disk clutches running in oil, which were much in vogue some years ago, trouble was encountered by warping or distortion of the plates. In traffic it is necessary to operate the clutch frequently and this raises its temperature and that of the oil, and as oil loses in viscocity when heated, the friction between the plates is greatly augmented; this induces a still higher temperature with the consequent warping of the plates.

The Hele-Shaw clutch runs partly submerged in oil. It is claimed by the makers that in this clutch the difficulties encountered with the ordinary straight disks running in oil have been

overcome. The forward end of the clutch shaft is supported in a ball bearing, centered in the bearing plate of the clutch. A steel core rides on this shaft and is driven by a square fit. This core (Fig. 104) has driving teeth on its exterior which engage with the inner plates made of steel. A bushed sleeve slides on the shaft and ends in a pressure plate, which controls the pressure between the inner and outer plates. This "presser" is normally kept tight against the plates by the pressure from the main clutch spring, the other end of this spring being held by an adjusting nut screwed into the boss in the end of the case. Therefore, no thrust is exerted against the engine shaft or gear box while the clutch is engaged. The spring adjusting nut is locked in position by a cotter-pin secured key, which can be readily disengaged to allow the spring pressure to be altered by this adjusting nut. When the full spring pressure forces the outer and inner plates together the power of the engine is transmitted to the gear shaft. By moving back the actuating-ring the presser is disengaged and the plates separated. This actuating-ring has two trunnions engaging with the pedal levers whereby the motion of the foot pedal is transmitted to the presser through the combined-thrust and radial bearing and its assembling lock-nut. The outer plates on their outside diameter are fitted with small flat springs which open the plate pack when disengaged.

In plate clutches running in oil, there is a slight drag when released, caused by the skin friction of the film of oil between the plates, which tends to keep the clutch shaft revolving when the clutch is out of gear. To throw the first speed or the reverse into mesh, it is preferable that this shaft should be stopped. Accordingly, the actuating ring terminates in a brake which presses against a small disk revolving with the gear shaft which constitutes a sufficiently powerful clutch brake. This brake is made adjustable to allow for wear of the plates. It is claimed that the points of excellence of the V-grooved twin plates are their uniform rate of engagement and their flexibility.

Clutch Brakes.—Various clutch brakes were described in the preceding pages. Their object is to stop the spinning of transmission gears when the clutch is disengaged, as this will render it easier to shift gears without clashing under certain conditions. A clutch brake is especially advantageous when the clutch driven parts are comparatively heavy or when their diameter is large, which is, in other words, when the inertia of the revolving

parts is great. The clutch brake, however, should not always be applied, but when stepping-up from a lower into a higher gear, it is useful to apply it, for this slows down the countershaft gear and the ideal condition for "shifting" is when both gears, (the sliding gear and the countershaft gear) are running at approximately the same peripheral speed. On the other hand, when shifting from a higher speed into a lower, (when "stepping-down") the sliding gear on the spline shaft, (which is permanently connected with the rear axle through the propeller shaft) has a higher peripheral speed than the countershaft gear, hence, in this case, no clutch brake is necessary; on the contrary, when coming through "neutral," reengaging the clutch for an instant and speeding up the engine, renders possible a perfect shift. For this reason, these brakes are so arranged as to be operative only when the foot lever is fully depressed, hence, its actuation is at the drivers, discretion. The clutch brake affects only the countershaft gears (see Fig. 159) since the main gear of the countershaft is in permanent mesh with the main driving gear and the latter is in permanent connection with the clutch driven parts through the main drive shaft.

Energy Stored in Rotating Parts.—The cone weight and diameter, in cone clutches, and the driven parts in disk or plate clutches, as well as the weight and diameter of the countershaft gears should be kept down to a minimum consistent with strength and other requirements, to reduce their inertia for the reasons previously mentioned. In Chap. XIII of Vol. I it was shown that the kinetic energy stored in a flywheel, or in any rotating mass is $\frac{WV^2}{2g}$ ft. lb. per second; W being the weight of the rotating mass, V the lineal velocity of the center of the rotating mass, and g the acceleration due to gravity (= 32.2 ft. per second per second). If V, which depends on the number of revolutions, on the mean radius of the clutch cone and on the radius of gyration of the rotating parts, is changed, either increased or decreased, a certain amount of energy has to be expended to bring about such change in velocity, and this energy is supplied by the transmission gears of the primary shaft, which is in permanent connection with the road wheels through the rear axle and propeller shaft. If the inertia force (see Chap. XII, Vol. I,) of the revolving parts is high, there will be a greater pressure between the teeth of the gears when they are being meshed. It was stated before, when dis-

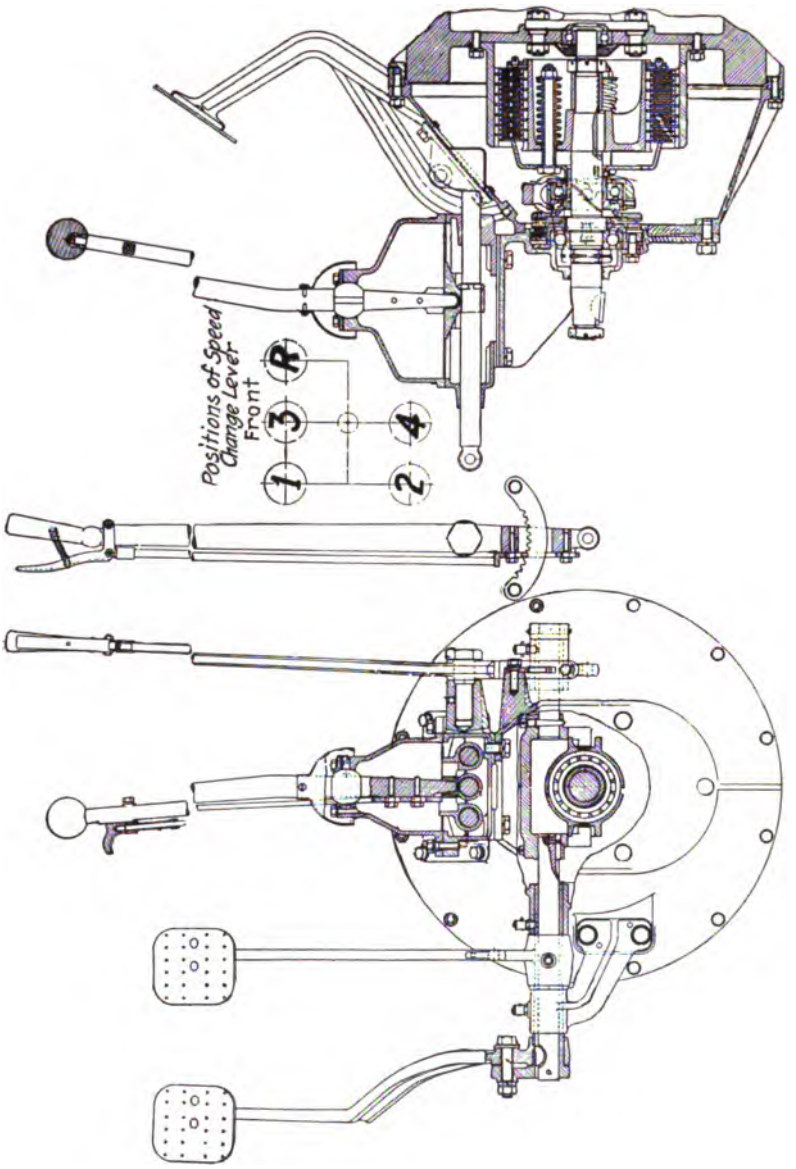


FIG. 105.—Pierce-Arrow 2-ton truck clutch.

cussing cone clutches, that the smaller the radius of the cone, the greater the required spring pressure for transmitting a certain torque, hence manufacturers endeavor to make the cone diameter large and its weight small. For this reason, cones are frequently made of aluminum or pressed steel.

Clutch Operating Mechanism.—The clutch is thrown out of engagement by a slight angular motion of the clutch throw-out shaft, or clutch release shaft, to which the throw-out yoke is attached, (see Fig. 93). Here the throw-out shaft *34* is actuated

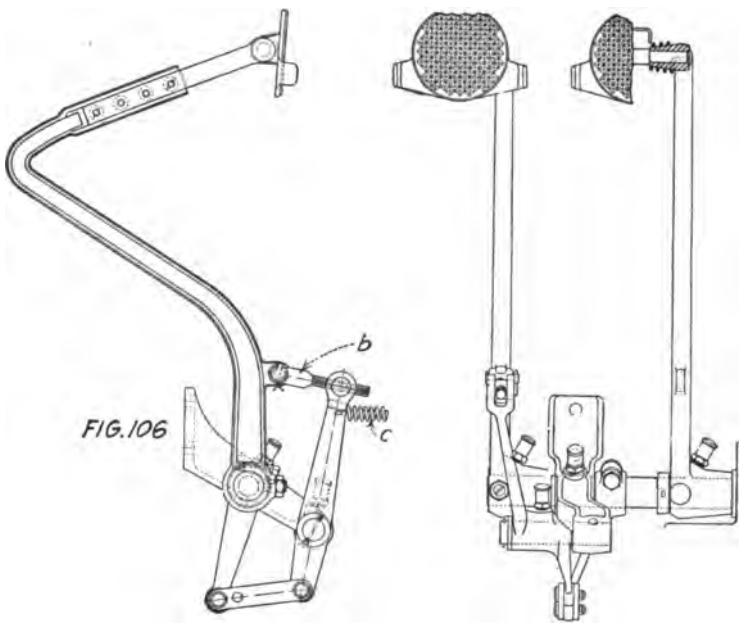


FIG. 106.—Peerless passenger car clutch foot levers.

by lever *38* which is keyed to it. The said lever is operated by the foot lever *45* which is pivoted on foot lever shaft *53*, and is connected with lever *38* by means of adjustable connections *41* and *42*. Sometime's, as in Fig. 105, (the Pierce-Arrow 2-ton truck clutch, levers and pedals) the foot levers are directly seated on the throw-out shaft. But even here, it may be noted, the foot lever is not rigidly attached to the shaft but to a small lever which is keyed to the shaft. By means of the bolt shown at *a* the position of the foot lever can be adjusted to the desired position. The 3½- and 5-ton truck made by the Pierce-Arrow Motor Car Co. are equipped with similar clutches.

Frequently the foot levers are not resting directly on the clutch throw-out shaft but on a separate shaft, and various means are provided for adjustments, both for the position of the lever as compared to the throw-out shaft, as well as for the height of the foot pads. Figure 106 shows the foot levers of the Peerless passenger car. By means of the clutch pedal link *b* the clutch is adjusted so that the clutch pedal may have a slight free movement before the clutch begins to release. A spring *c* keeps the

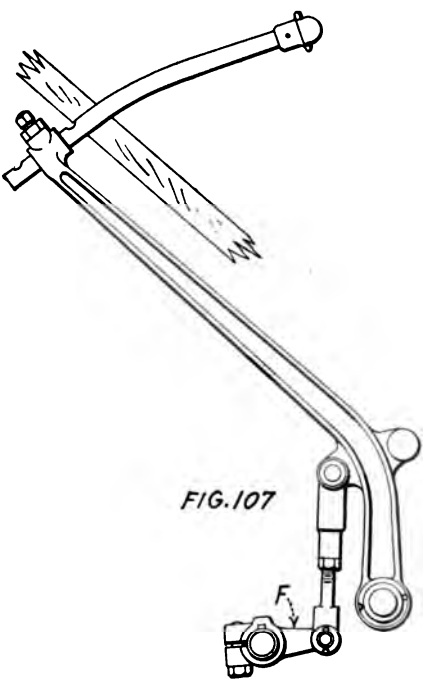


FIG. 107.—Clutch pedal adjustment of King car.

foot lever in its upright position that the weight of the lever may not rest on the throw-out yoke of the clutch, thereby avoiding a constant load on the clutch thrust bearing, *i.e.*, this bearing will not be in continuous action. On the other hand, in some of the models the clutch throw-out bearing is always in operation. Figure 107 shows the clutch pedal adjustment of the King car, *F* being in this case the clutch release shaft. Figure 108 is the clutch lever adjustment of the Oakland car and is self explanatory.

The foot lever linkage is usually so proportioned that a pedal pressure of about 40 lb. will operate the clutch. As an example we will take the clutch of the White truck, (Fig. 93). Suppose it required a force of 300 lb. at the end of the prongs of yoke *36* to operate the clutch. (This yoke in practice is not in the position shown in the illustration but it is situated with the prongs vertical, *i.e.*, the shaft *34* is either directly above or below the clutch pull-out shoe *29*.) If the operative length of the prongs is 3 in. as indicated, then $300 \times 3 = 900$ lb.-inches which is the turning moment necessary on shaft *34* to operate the clutch.

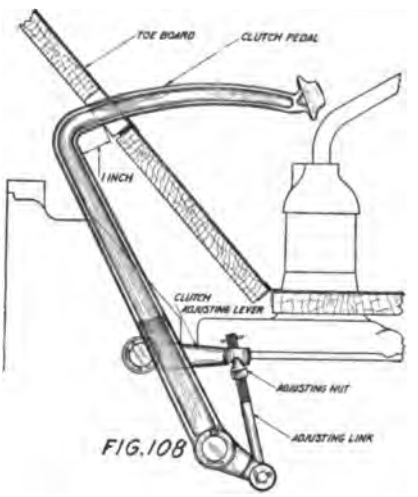


FIG. 108.—Clutch lever adjustment of Oakland car.

If the lever *38* (which is attached to shaft *34*) is 5 in. long and placed vertically as shown, the force at its end or axis *43* is $\frac{900}{5} = 180$ lb., this of course will also be the pressure at pin or axis *43* (provided the rod *41* is substantially at right angles to lever *38*). If the distance from axis *43* to the center of the foot lever shaft is 4 in., the twisting moment of this shaft is $4 \times 180 = 720$ lb.-inches. In order to have 40 lb. pressure at the foot pedal the lever must have a definite length x. Continuing the calculations as before we have $\frac{720}{X}$ foot pressure $= 40$ lb. Hence $X = \frac{720}{40} = 18$ in. These calculations only hold good as long as the levers are at right angles to the rods; in practice, this

is substantially the case. Looking at the lower half of the foot lever, we notice that it is not at right angles to rod *41*, however, when the clutch is operated and the pedal moved down, it will assume an angle of approximatly 90° with rod *41*. On the other hand should it be necessary to have an angle greatly differing from 90° then, instead of the length of the lever, the distance from the rod (and at right angles to it) to the axis of the lever must be taken, (see Fig. 109).

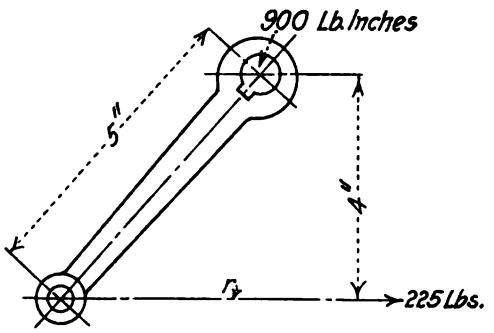


FIG. 109.

If the shaft is under a twisting moment of 900 lb.-inches as before, and the lever 5 in. long, and the distance from rod *r*, as indicated, to the axis of the shaft 4 in. on account of the angularity of the lever, then the force exerted through the rod is not $\frac{900}{5}$ as before, but $\frac{900}{4} = 225$ lb. The force or pull at the end of the levers is always expressed in pounds, while the shafts sustain twisting moments which are expressed in pound-inches.

CHAPTER V

SHAFTS AND UNIVERSAL JOINTS

The stresses produced in a shaft which is rotated or twisted are called torsional stresses; these stresses produce chiefly shearing stresses in the shaft, hence torsional stresses may be termed shearing stresses.

In Vol. I of this series, Chap. XVII, it was shown that the modulus of elasticity, $E = \frac{\text{stress}}{\text{strain}}$; as long as the elastic limit of the material is not exceeded, the ratio of the stress to the strain (when the stress in in tension the strain is the actual elongation and when in compression, the reduction per inch of length) is constant and is called the coefficient of elasticity or more frequently the *modulus of elasticity*. When the material is subjected to a shearing stress, (arising from torsion, for example) it will likewise be strained (deformed). If, when the material is so strained by torsion, the elastic limit is not exceeded, the strain (in a shaft, the circumferential deformation per inch of shaft length) will be proportional to the stress, and $\frac{\text{stress(shearing)}}{\text{strain}} =$ *modulus of elasticity in shear* (sometimes called the modulus of transverse elasticity or the coefficient of rigidity). This shearing modulus of steel is approximately 12,000,000 (lb. per square inch).

Figure 110 shows a shaft and imagining for instance one end fixed, so it cannot turn, and attached to the front end a gear wheel K which we endeavor to turn by another gear wheel N. The force applied to the gear wheel K, is, in this case, the resultant of the pressure P between the teeth, and this force or pressure is exerted at a distance d from the center of the shaft; the product dP that is to say, the distance d multiplied by the force P, is the moment of the force tending to produce rotation, and this moment is called the *twisting moment* or torque T, and it is expressed in pound-feet or pound-inches, depending on whether d is given in feet or in inches, thus $T = dP$. If for instance d is 10 in., and P is 200 lb. the twisting moment is $10 \times 200 = 2{,}000$ lb.-inches.

Suppose, before trying to turn the wheel, we mark straight small lines *abc*, at the end of the shaft, from the center *O* outward. When an effort is made to turn the shaft, by means of gear wheel *K*, and a certain pressure or force is exerted on the teeth of the gear wheel, the small lines will be displaced to positions, $a_1b_1c_1$, but they will remain in line from the center *O* outward, as shown, so long as the elastic limit of the material is not exceeded. While at the outer circumference of the shaft the end of the small line *a* is displaced to a comparatively large distance, to a_1 which shows that the stress here is large, this distance becomes less

FIG. 110.

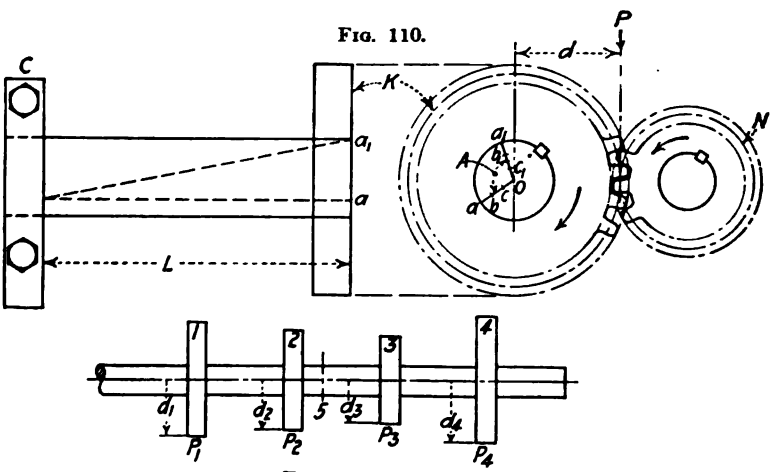


FIG. 110a.

FIGS. 110 and 110a.—Illustrating shaft stresses.

as the line runs toward the center, consequently the stress must become smaller (from *c* to c_1, the distance is very small) and at the center there is no stress at all. Hence *the stress* in the layers or fibers of *a shaft varies in direct proportion to their distance from the shaft center*, as long as the elastic limit of the material is not exceeded.

On page 13 the shearing strength of various materials is given. The shear or transverse elastic limit and the modulus of elasticity in shear are less than those given for tension.

For steel, this modulus in shear is only about 40 per cent of the modulus of the material when under tension and the transverse elastic limit is about 35 per cent of the shearing strength of the metal. The material, and the size and shape of the shaft, will

resist torsion, and the ability of the shaft to resist torsion is the *resisting moment* of the shaft and this equals the twisting moment T. On page 50 the polar moment of inertia was defined; this polar moment of inertia is denoted by J (to distinguish it from the moment of inertia of an area I) and it was shown that $J = Av^2$. If $x = \frac{1}{2}D$, the distance from the center of the shaft to its outer periphery (the radius, in the case of a round shaft), then $\frac{J}{x}$ is, so to say, the measure of the strength of the shaft, and this is called the polar section modulus Z_p of a section.

For a circular solid shaft,

$$J = \frac{\pi D^4}{32} = .0982\ D^4 \text{ (where } D = \text{diameter of shaft)}$$

$$Z_p = \frac{J}{x} = \frac{\frac{\pi D^4}{32}}{1/2\ D} = \frac{\pi D^3}{16} = .1963\ D^3$$

For a hollow shaft with an internal diameter d, and external diameter D,

$$J = \frac{\pi(D^4 - d^4)}{32} = .0982\ (D^4 - d^4)$$

$$Z_p = \frac{\frac{\pi(D^4 - d^4)}{32}}{1/2\ D} = \frac{\pi(D^4 - d^4)}{16\ D} = .1963\left(\frac{D^4 - d^4}{D}\right)$$

For a square shaft where D is the distance parallel to a square side, $Z_p = .21D^3$.

The twisting moment $T = S_s Z_p$, where S_s is the maximum shearing stress, that is, the stress on the outer periphery of the shaft.

Hence $S_s = \frac{T}{Z_p}$, and $Z_p = \frac{T}{S_s}$. With the aid of this formula the stress in the metal of a shaft can be determined when the twisting moment T is known. It might be stated that the twisting moment T at any section of a shaft is the algebraic sum of the moments of all the forces on either side of the section. For instance in Fig. 110*a*, imagine the shaft driven by gear wheel *2*, and the pressure or force exerted at distance d_2 to be P_2. If $d_2 = 6$, and $P_2 = 2{,}000$ lb. then $T = d_2P_2 = 6 \times 2{,}000 = 12{,}000$ lb.-inches.

Supposing the other three gears to be in mesh with gear wheels driving various machines or units. Evidently the three gears, *1*, *3* and *4*, together, absorb the power delivered by gear wheel *2*,

hence the twisting moments of those three gears must equal 12,000 lb.-inches. Let us assume

$d_1 = 7$ in., $P_1 = 400$ lb.; the twisting moment $T_1 = d_1P_1 =$ 2,800 lb.-inches.
$d_3 = 4$ in., $P_3 = 2{,}000$ lb., $T_3 = d_3P_3 = 8{,}000$ lb.-inches.
$d_4 = 8$ in, $P_4 = 150$ lb., $T_4 = d_4P_4 = 1{,}200$ lb.-inches.
$T_2 = T_1 + T_3 + T_4 = 12{,}000$ lb.-inches.

Example.—Find the twisting moment in a section of the shaft at dotted line *5*. Considering the twisting moments at the right of this section, we have $T = T_3 + T_4 = 8{,}000 + 1{,}200 = 9{,}200$ lb.-inches. We might also take the twisting moments on the left of the section, here we have $T_2 = 12{,}000$ lb.-inches and $T_1 = 2{,}800$, but the moments of T_1 is in opposition to T_2, for the former is the driving gear, and the latter the driven gear, hence $T = T_2 - T_1 = 12{,}000 - 2{,}800 = 9{,}200$ lb.-inches.

If desired, the twisting moments may be denoted as positive and negative, the same as was explained under bending moments, Chap. II. As an example, find the maximum shearing stress S_s in the shaft when the diameter D is 1.5 in. and the maximum twisting moment 9,200 lb. It was shown before that $T = S_sZ_p$. To find the diameter of the shaft for a given stress S_s, we solve Hence

$$S_s = \frac{T}{Z_p}.$$

For solid shaft 1.5 in. in diameter,

$$Z_p = .1963\ (1.5)^3 = .662,$$

$$\text{therefore } S = \frac{9{,}200}{.662} = 13{,}900 \text{ lb. approximately.}$$

D from the formula given before; $T = S_sZ = S_s \times .1963^3$, hence $D = \sqrt[3]{\frac{T}{S_s \times .1963}}$; If T is 9,200 as before, and S_s is to be 13,900 then

$$D = \sqrt[3]{\frac{9{,}200}{13{,}900 \times .1963}} = \sqrt[3]{3.375} = 1.5 \text{ in.}$$

To find the equivalent strength of a hollow shaft, d_1 being the inside and D_1 the outside diameter, we have the formula

$$D^3 = \frac{D_1^4 - d_1^4}{D_1}$$

If we now pre-determine the outside diameter D_1 and wish to find d_1, we solve for d_1, from the last equation $d_1^4 = D_1^4 - D^3 \times D_1$. If D_1 is 2 in., then the hole d_1 in the shaft (keeping the strength of this shaft the same as the solid shaft 1.5 in. in diameter) will be $d_1^4 = 2^4 - 1.5^3 \times 2 = 16 - 3.375 \times 2 = 9.25$, and $d_1 = \sqrt[4]{9.25} = 1.745$ in.

In other words, if the diameter of the shaft is increased from 1.5 in. to 2 in. it may be bored out almost 1¾ in. in diameter and still have the same strength as before, but the weight would be considerably less. Regarding the strength of a shaft it might be stated, that if it is made tubular or hollow, by boring out the center, so that the area of the section bored out is one-fourth the area of the whole section, the strength is reduced ¼ × ¼ = 1/16; if one-half of the area is bored out it will be weakened only ½ × ½ = ¼, thus still retaining 75 per cent of its original strength, while its weight is reduced to one-half. The shearing strength of steel is about 85 per cent of its tensile strength and the elastic limit when in shear is only about 35 per cent of its ultimate shearing strength. If it is desired to find the maximum strength of a shaft in torsion, it may be derived from the formula $T = S_s Z_p$, where S_s is the ultimate shearing strength of the material.

For instance, suppose S_s of a certain alloy steel is 130,000 lb. per square inch, and the shaft diameter is 1.5 in. as before, then this shaft would rupture when $T = 130{,}000 \times .1963\ D^3 = 86{,}000$ lb.-inches. But the elastic limit in shear is reached when S_s is about $130{,}000 \times .35 = 45{,}000$ lb. per square inch since the elastic limit in shear is about 35 per cent of the shearing strength. When there is a combined twisting moment T, and a bending moment B, the stress is the metal will be greater than that due to either of the two moments alone. If T_e is the twisting moment which would create the same stress in the shaft as that due to the combined twisting moment T and bending moment B, then we may determine T_e, usually called equivalent twisting moment, from the formula, $T_e = \sqrt{B^2 + T^2}$.

After T_e has been determined in pound-inches we may proceed to find the stresses or the size of the shaft from the formula given before, replacing T by T_e, thus $T_e = S_s Z_p$.

(See page 131 for the radial pressure in the center of a shaft arising from the gear teeth pressure. This radial pressure is analogous to W in the beam formulae, Chap. II.) Sometimes it is necessary to know the angle of torsion A (see Fig. 110); as a rule

this is not necessary in shafts used in motor vehicles, but it might be stated that the longer the shaft L, the greater will be the angle for a given twisting moment. If E_s is the modulus of elasticity in shear (about 12,000,000 for steel), L, the length of the shaft between the driving wheel and the driven wheel, D the shaft diameter, and A the angular displacement in degrees, then $A^0 = \frac{583.7TL}{E_sD^4}$, for solid shafts, and $\frac{583.7TL}{E_s(D_1^4 - d_1^4)}$ for hollow shafts.

This angular displacement can be determined when the twisting moment T is known.

The angle A should never be more than .1° per foot of shaft length; in shafts used in motor vehicles it is usually very much less.

Horsepower Transmitted by Shafts.—In Vol. I, Engines, and in the preceding chapter on clutches, it was shown that the brake horsepower (hp.) of an engine was hp. $= \frac{2\pi TR}{33{,}000}$, (where T_1 the torque $= dP$), and if the torque T is given in pound-inches, hp. $= \frac{2\pi TR}{33{,}000 \times 12} = \frac{TR}{63{,}025}$, where R is the number of revolutions per minute.

From the above formula the torque or twisting moment $T = \frac{\text{hp. } 63{,}025}{R}$. Hence, when the speed and hp. are known, the twisting moment of the shaft may be determined, and since we also have $T = S_sZ_p$, we can determine the maximum power the shaft can transmit. $T = S_sZ_p = S_s \times .1963D^3$, (see page 119); hence for a solid shaft, $S_s \times .1963\ D^3 = \frac{\text{hp. } 63{,}025}{R}$, and the power the shaft can transmit hp. $= \frac{S_s \times .1963\ D^3 \times R}{63{,}025} = \frac{S_sD^3R}{321{,}000}$, from which $D^3 = \frac{\text{hp. } 321{,}000}{S_sR}$.

For a hollow shaft we have

$$S_s\,Z_p = S_s \times .1963 \left\{\frac{D_1^4 - d_1^4}{D_1}\right\} = \frac{\text{hp. } 63{,}025}{R} \text{ and}$$

$$\text{hp} = \frac{S_s \times .1963(D_1^4 - d_1^4)R}{63{,}025\ D_1} = \frac{S_s(D_1^4 - d_1^4)R}{321{,}000\ D_1}.$$

By the aid of these formulae the power a shaft is able to transmit, before rupture, is readily found when the shearing strength S_s of the material is known and is substituted for S_s in the above

formula, while if we take S_s as the safe working stress in the metal (under normal load) say, for instance 10,000 lb. per square inch, and substituting this value for S_s in the equation, we obtain the hp. the shaft will transmit normally without overstraining the metal. For instance, if the ultimate shearing strength of the metal is 80,000 lb. per square inch, and the working stress is 10,000 lb. per square inch, then we would be working with a factor of safety of $\frac{80,000}{10,000} = 8$.

However, the elastic limit in shear of this metal is only (35 per cent of its shearing strength) $80,000 \times .35 = 27,000$ lb. per square inch, and the factor of safety with respect to the elastic limit is only $\frac{27,000}{10,000} = 2.7$. The author recommends that the elastic limit be considered when determining shaft dimensions, as it is a safer indication of its strength.

Whirling of Propeller Shafts.—When a shaft is supported between two bearings there is a tendency for it to bend or sag in the center. As it rotates, it will continuously bend to-and-fro, depending in amount, on the length of the shaft, its diameter, and on the speed of its rotation, and if this bending action occurs near the natural period of vibration of the shaft, a harmonic motion will be established, which will cause the shaft to whip tremendously as it rotates, and this whipping action is called "Whirling." Whirling will occur at certain speeds; if the speed is exceeded or diminished, the whirling will stop.

If R is the number of revolutions per minute when whirling takes place; d the diameter of the shaft in inches and L the length of the shaft between the bearings in feet, then, according to Professor Dunkerly (see GOODMAN, "Mechanics"), the speed R at which whirling takes place can be found from the formula, $R = 32,864 \frac{d}{L^2}$, and $L = 181\sqrt{\frac{d}{R}}$. This formula is for solid shafts supported in free bearings at their ends as used in motor vehicle propeller shafts, where the shaft is subjected to torsion only. Hence, by the aid of the formula, we can proportion the length and the diameter so that whirling is avoided.

Universal Joints.—In this country, universal joints are generally made by manufacturers who specialize in their production. For this reason only a few salient points will here be discussed.

Universal joints are necessary between a driving shaft and a

driven shaft, if any angularity exists between them. In motor vehicles, when the springs are flexed or when stresses in the frame cause it to deflect, a misalignment of the shafts usually occur. Tests made in the laboratory of the University of Kansas (see A. LUDLOW CLAYDEN, S.A.E. Trans., 1916, Part 2), disclosed that the efficiency of universal joints is very nearly 99 per cent for angular deflections up to 7° or 8°; this efficiency decreasing more rapidly as the angle increases. The angular deflection A refers to the angle of deflection of one shaft with respect to the axis of the other, (see Fig. 111).

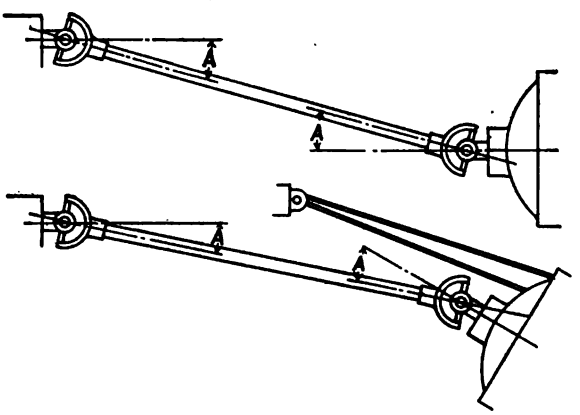


FIG. 111.—Propeller shafts.

If there is one universal joint between two shafts and if the driving shaft runs at a uniform speed, the driven shaft will fluctuate in speed, (in each revolution it will twice be higher and twice lower than the driving shaft) the amount of speed variation depending on the angle of deflection A. Professor Rankine has shown that if v is the angular velocity of the driving shaft; v_1 the minimum and v_2 the maximum angular velocities of the driven shaft, $v_1 = v \cos A$, and $v_2 = \dfrac{v}{\cos A}$.

The maximum difference in speed between the driving and the driven shafts is about 2 per cent when the angle of deflection is $7\frac{1}{2}°$ and about 7 per cent when this deflection is 15°. Evidently, if there is only one universal joint between the transmission gear shaft and the bevel gear pinion shaft of the rear axle, the latter will tend to vary in speed during each quarter revolution of the

former and will attain two maximum and two minimum values in each revolution; this must necessarily cause an extra strain in the shafts and gears. However, inasmuch as a number of cars are successfully running with one universal joint, it proves, that, as long as the angle of deflection is not too large, one universal joint is practicable. On the other hand, by using two universal joints in the propeller shaft, one near the gear box and one near the rear axle, this speed fluctuation between the transmission shaft and rear axle can be eliminated. It will not be eliminated in the propeller shaft proper, between the two joints, but since this shaft is comparatively light, it is unimportant.

When using two universal joints, it should be remembered that the two forks, one at each end of the propeller shaft, must be set parallel to each other (not at right angles), in order to compensate for the speed fluctuations in the joints.

Figure 111 makes this plain and it also shows that the angular deflection should be the same in the front as in the rear of the propeller shaft. There are two methods for accomplishing this, (*a*) shows the two shafts (transmission and bevel pinion) parallel, while (*b*) shows them set so that the angle between the shafts is decreased, and this appears to be a more advantageous construction for this reason. Of course, it should be kept in mind that the angle will change when the springs are deflected and it is suggested that the engineer lay out carefully the design, with the springs fully depressed and in their "no-load" position, and then set the angle of the two joints so that they will run uniformly most of the time.

CHAPTER VI

TRANSMISSION OR CHANGE SPEED GEARS

The speed of an internal combustion engine can be varied within certain limits and this renders possible a certain change in the speed of the motor vehicle. It was shown (see Vol. I, Engines) that the power of an engine, is, in a measure, proportional to its speed of rotation. At low engine speed therefore, the power generated would not be sufficient to propel the motor vehicle slowly, when slow travel is necessary as a certain minimum power is required for its propulsion. Likewise, when encountering steep hills, the power of the engine would not be sufficient if it were attempted to drive the car up on the incline at the same speed as on the level. When the power required is greater than the engine can deliver, it will at first slow down and as this reduces the engine power still further, it will finally stop. For these reasons some form of changeable speed gearing is necessary that the car speed may be changed without altering the engine speed. In other words, the ratio of the number of revolutions of the engine to that of the driving wheels must be made variable. This is accomplished by having a number of gear wheels of various sizes, in a box, called the gear box, with means for meshing certain pairs of gears.

By far the most common type of transmission gear in use is the selective sliding gear, where any pair of gears selected may be meshed, by sliding one of the pair sidewise and into mesh with the other.

Gear Box Location.—In more than 85 per cent of the passenger car models the unit power plant is used, where the gear box is in one unit with the clutch housing, the latter being a unit with the crankcase, approximately 10 per cent of the models have the gear box or gearset placed amidship, that is, some distance behind the clutch, or about in the middle of the chassis. On the other hand, in the number of trucks on the market, about 50 per cent of the models have the gearset located amidship while the others employ the unit power plant construction.

The latter type offers some advantages as to cost and simplification in that the universal joint and the shaft between the clutch and the gear box are dispensed with and that it is not necessary to align the transmission with the engine; its disadvantages are chiefly lack of accessibility. In some models embodying the unit power plant it is most difficult to get at the clutch to renew worn parts, or make adjustments. The practice of locating the gear box at the rear axle or at the rear end of the torque tube is very limited; it has the disadvantage of increasing the unsprung weight of the rear axle. Its advantage is that a lighter propeller shaft and lighter universal joints can be employed since their speed is higher and the torque is less. The torque increases with each speed reduction. The amidship location has the advantage of distributing the weight over a larger portion of the chassis.

Number of Speeds.—In passenger cars, with very few exceptions, three speeds and reverse is the standard practice, while in the number of truck models on the market, over 60 per cent have a four speed transmission (four speeds forward and one reverse). This type has been on the increase lately, as it affords more truck flexibility, enabling the driver to mount inclines at the most efficient motor speed.

Gear Ratios.—The gear ratios employed in the different passenger car models vary greatly, but usually the low gear is between .27 and .30 (27 to 30 per cent) of the high speed or the direct drive, which equals 1/.27 to 1/.30, thus from about 3.7 to 1, (1/.27 = 3.7) to 3.33 to 1; the second speed or the intermediate gear is about 60 per cent of the high, or 1/.60 = 1.66 to 1. On the high gear the drive is direct, *viz.*, no reduction is made in the gear box, the engine shaft being merely connected direct to the propeller shaft. When four speeds are employed, the third is about .70 or 70 per cent of the high, which is 1/.70 = 1.43 to 1. Sometimes, in four speed transmissions, the third is directly connected, while the fourth is over-geared, for instance 1 to 1.25. In trucks the first gear as well as the other gear ratios are frequently made higher than given above, especially in the heavier models, and often the reverse is from 20 to 30 per cent higher (a higher ratio means a lower speed) than the low gear. In trucks, the ratio of the propeller shaft speed to the rear axle speed is considerably higher than in passenger cars to obtain an increased torque and slower speed of the wheels.

In the Class B truck transmission (see Fig. 112), the main driv-

ing gear *4*, on the drive gear shaft *G*, has 19 teeth, and the driven gear *5*, of the countershaft *C*, 41 teeth; on the low gear or the 1st speed, the driver *1* on countershaft, has 16 teeth, the driven gear *I*, 44 teeth; 2nd speed, the driver *2* on countershaft, has 24 teeth, the driven gear *II*, 36 teeth; 3rd speed, the driver *3* on countershaft, has 33 teeth, the driven gear *III*, 27 teeth; 4th speed is direct, the gear *III* being shifted to the left when its teeth *IV*, cut internally, mesh with teeth of gear wheel *4*. In this case the teeth form simply a positive clutch and are locked together, thereby locking main drive shaft *G* with primary shaft *S*.

The reverse speed in this transmission has the same ratio as the low speed; the latter being sufficiently low, much more so than is usually the case, to climb any hill, at full load, under the worst conditions.

On the first gear, the proportion of the speed of shaft *G* is to the speed of countershaft *C*, as 41 is to 19, or $41/19 = 2.158$ times the countershaft speed *C*, and speed of *C* is to *S* as $44/16 = 2.75$. Hence the speed of *G* is to speed of *S* as $41/19 \times 44/16 = 2.158 \times 2.75 = 5.93$ times the speed of *S*, or as usually termed, 5.93 to 1, which is called the low gear ratio. A rule for finding the gear ratio of the clutch gear shaft to the reduced speed of the transmission shaft (often called the sliding gear shaft, main shaft or primary shaft) *S* is: Divide the product of teeth of driven gears by the product of the teeth of the driving gears.

On the second speed the ratio is $\frac{41 \times 36}{19 \times 24} = 3.23$ to 1. On the third the ratio is 1.76 to 1. The fourth speed is direct, 1 to 1. In designing transmission, it must be remembered that the number of teeth in each pair of gears (of both together) must be alike, in order to have the same distance between their centers.

Transmission Gears.—Before finding the bearing pressures in the various bearings of a change speed gear box, it is necessary to examine the gears in use, (see also chapter on Gearing) their shape and design, since the shape of the tooth will have an influence upon the bearing pressure. Two types of teeth have formerly been in common use, the epicycloidal and the involute; at the present time only the involute type is in general use, either with the standard tooth or the stub tooth form, the latter having first made its appearance in 1898 and is now the most common type used in the motor vehicle industry. With the involute type of

gear, the outlines of the working parts of a tooth are single curves, and the pressure angle (the line of contact or the line of action), between the driver and the driven gear teeth, is in the standard tooth, 14½° from a line drawn tangent to the pitch circles of the two gear wheels, while with the stub tooth the pressure angle is 20°.

With the epicycloidal system, the outline of the tooth has a double curve, the formation of the curve changing at the pitch circle. The advantage of involute teeth is, that in action, they may be separated a little from their normal positions without interfering with the angular velocity, which is not possible with any other kind of tooth. Other claims made for the 20° involute or stub-tooth are: It reduces the sliding action and increases the rolling action between the tooth surfaces and hence there is less wear than in the standard 14½° tooth; the tooth is shorter and wider at the base and therefore has an increased strength, which increase in small pinions is very high. On account of this greatly increased tooth strength in small gear wheels, a smaller pitch may be used for such gears, thus increasing the number of teeth in contact. In the Class B transmission, the teeth are 20° involute, as will be described later.

Bearing Pressure Calculations.—In motor vehicles, the stub-tooth is most generally used in this country. It was stated before that with this gear there is, between the teeth, a pressure angle of 20°, to which must be added a friction angle of 3°, making a total of 23° (see Chapt. XV Gears). When one gear wheel drives another, there is a certain amount of friction between the teeth; this friction increases the angle, or the resultant direction of the pressure line between the teeth by about 3°. In practice, this angle due to the friction, is added to the pressure angle. In spiral spur gears or spiral bevels, there is more of a rolling action than in straight spur or bevel gears, and for this reason no friction angle, but only the pressure angle, is considered with spiral gears.

In (Fig. 112), which is the transmission of the Class B truck, the pinion *4* is the driving gear which constantly meshes with gear *5* of the countershaft. The problem is to find the bearing pressure when the various gears are in mesh. It might be stated that in determining bearing pressures, for gear boxes having three speeds forward, only the intermediate speed is considered, as practice has shown that bearings satisfying this load are ordinarily ample,

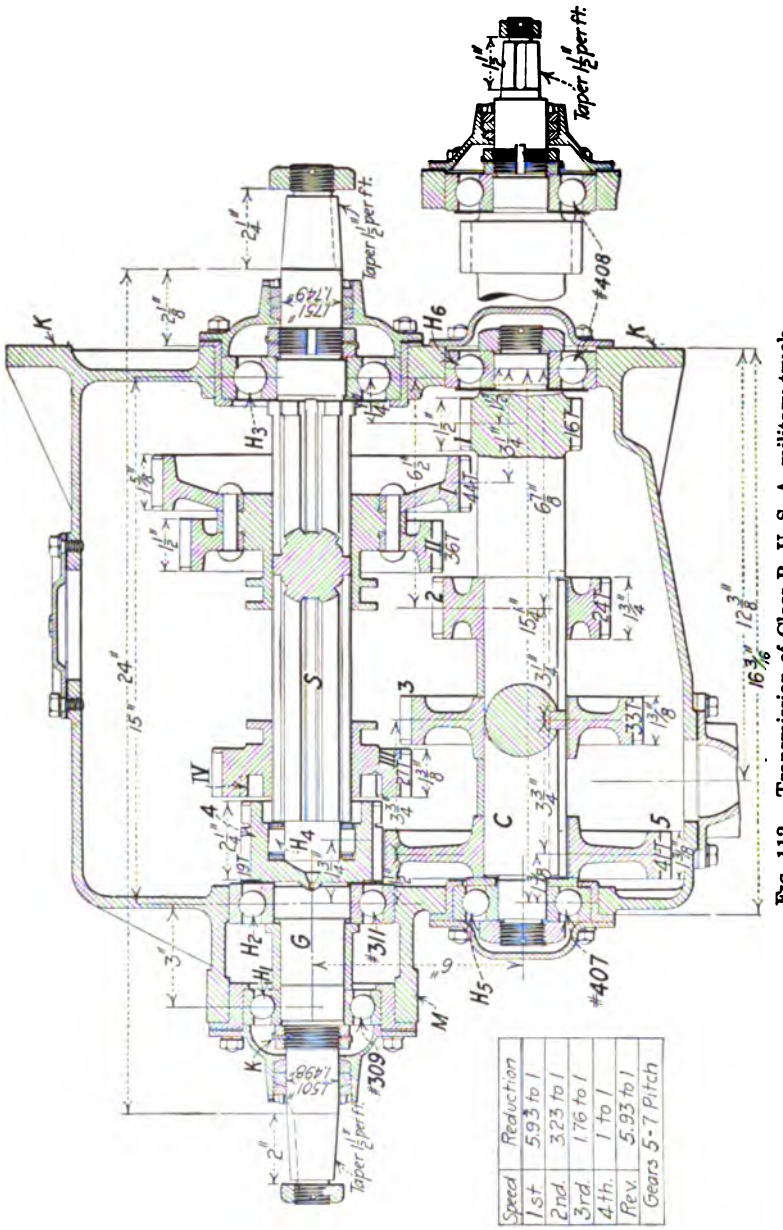


FIG. 112.—Transmission of Class B, U. S. A. military truck.

even though the bearing pressures may be greater at low gear and reverse, but the motor vehicle is, as a rule, not driven at these speeds very much. In four-speed gear sets, the second speed may be used for determining the bearing pressures.

In (Fig. 113), the pitch circle of the two gears, *4* and *5*, which are constantly in mesh, are represented diagramatically; *4* being the driving pinion directly connected to the crankshaft through the clutch, and *5* is the countershaft gear with which it is in continuous mesh. The former has 19 teeth, and the latter 41, as shown on the drawing. It was stated before that the Class B engine has a maximum torque of 2,800 lb.-inches. (See chapters on Torque Vol. I); the pinion has a pitch diameter of 3.8 in. (or a pitch radius of 1.9 in.) and torque $T = dP$, (d being the distance from the center of the gear wheel to the pitch circle having pressure P) *i.e.*, it equals the pitch radius d multiplied by the pressure P between the teeth, acting at a tangent, *viz.*, at right angles to line *A-A*, drawn through the center of both gears. Since $T = dP, P = \frac{T}{d}$. As was mentioned, T = 2,800 lb.-inches, consequently, $P = \frac{2{,}800}{1.9}$ = 1,474 lb.

If it were not for the pressure angle and the friction angle (in the stub-tooth 20° +3° = 23°) the radial pressure imparted to the center of the gear wheel would be 1,474 lb. To show that this is so, take (Fig. 114), where the wheel has a tendency to turn by the action of a weight of 100 lb. attached at the left, as shown. The pawl *b* resists the turning effort of the wheel which tends to move *b* upwards, as shown dotted, with a force of 100 lb. at the tooth, but in so doing the reaction of the pawl *b* exerts a downward pressure of 100 lb. on the one tooth shown on the wheel. Therefore, the bearing pressure in the center of the wheel would be 100 + 100 = 200 lb. the same as if a weight of 100 lb. where attached at each side of the wheel. If the weight shown is eliminated, and an effort is made to turn the *shaft*, with the same force, so that a pressure of 100 lb. is exerted on the pawl *b*, the pressure on the tooth shown would still be 100 lb. but the radial pressure (the reaction at the center of the wheel) would not be 200 lb. as before, but only 100 lb., since we are trying to turn the gear by its shaft. In other words, the pressure P of 1,474 (see Figs. 113 and 115)

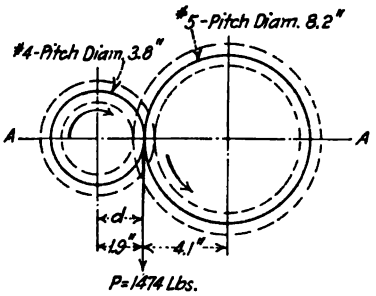


FIG. 113.

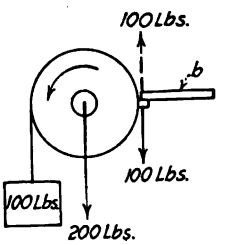


FIG. 114.

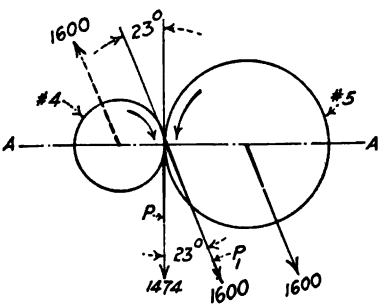


FIG. 115.

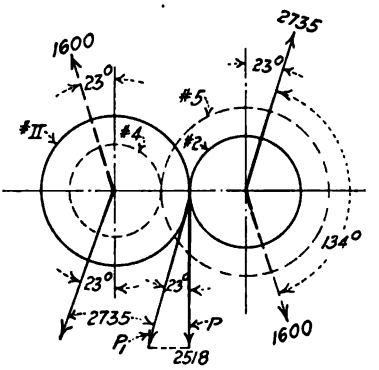


FIG. 117.

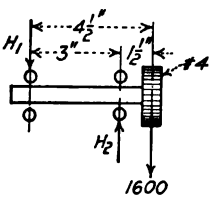


FIG. 116.

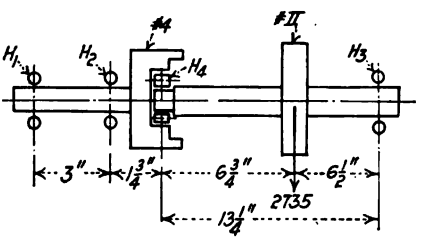


FIG. 118.

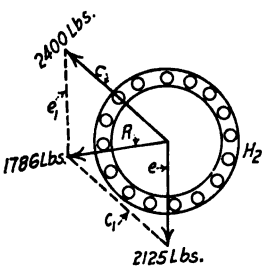


FIG. 119.

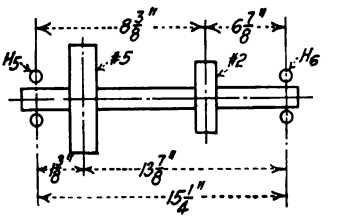


FIG. 120.

FIGS. 113 to 120.—Bearing loads in transmission.

will act tangentially to the gears (see Tangent Vol. I.) *i.e.*, at right angles to line *A-A*.

The pressure angle and friction angle will deflect the direction of the pressure on the teeth when gear wheel *4* drives *5* from the line at right angles to their common center, towards a new direction as indicated in (Fig. 115). However, the pressure or the force arising from the torque, at right angles to line *A-A*, is still 1,474 lb. but the pressure in the center of the driven gear wheel *5* will be deflected 23° due to the pressure from the teeth. If the pressure from the teeth, acting at an angle, induces pressure P (1,474 lb.) straight down, the actual pressure between the teeth P_1 will be greater than P, and can be found by simple trigonometry as follows: The length of line P divided by length of line P_1 equals the cosine of the angle of 23°. The cosine of 23° = .9205, hence, cosine 23° = $\frac{P}{P_1}$ = $\frac{1,474}{P_1}$ = .9205, and $P_1 = \frac{P}{\cos 23°} = \frac{1,474}{.9205}$ = 1600 lb. approximately. Therefore, the radial pressure against the center of the gear wheel *5* is 1,600 lb. (the same as the actual tooth pressure), and its direction is parallel to P_1 or 23° from the vertical. Since action equals reaction, evidently, the same pressure of 1,600 lb. is exerted against the center of the small gear (clutch gear), parallel to P_1, but in opposite direction thereto, as shown by the dotted line. Knowing the pressure toward the center of the gear wheel or from the center outward, the next step is to find the bearing pressures of the shaft on which the gear wheel is seated.

The division of the load between the two bearings, arising from the pressure on the gears, can be figured by the principle of moments (see pages 25 to 35). For instance, the pressure imposed on pinion *4*, amounting to 1,600 lb., is resisted by bearings H_1 and H_2, (see Figs. 116 and 112). The distance between the center of H_2 and the gear is 1½ in. as shown, and between H_1 and H_2, 3 in.

Taking moments about H_1, we have $4\frac{1}{2} \times 1,600 = 3 \times H_2$, hence $H_2 = \frac{4\frac{1}{2} \times 1,600}{3}$ = 2,400 lb., and taking moments about H_2 we write, $1\frac{1}{2} \times 1,600 = 3 \times H_1$, therefore $H_1 = \frac{1\frac{1}{2} \times 1,600}{3}$ = 800 lb. However, there will be an additional pressure on these bearings arising from load on bearing H_4,

(see Fig. 112), because another bearing is inside of gear *4*. In our case, if the second speed is considered, gears *2* and *II* will also be in mesh; the former being the driver on the countershaft, which drives gear *II*. In (Fig. 113), it was shown that the tangential pressure at pinion *4*, arising from the engine torque, is 1,474 lb. Therefore, the torque on the countershaft is 1,474 × pitch radius of gear 5 = 1,474 × 4.1 = 6.043 lb.-inches. In other words, the torque of the countershaft is 6,043 lb.-inches, regardless as to whether the first, second, third or the reverse speed gears are in mesh. The gear *2* (which drives gear *II*) has a pitch radius of 2.4, hence the tangential pressure at the pitch line is $\frac{6,043}{2.4}$ = 2,518 lb. (See Fig. 117.)

The gear *2* on the countershaft being now the driver, the pressure angle (plus the friction angle) is 23° as shown, and the actual tooth pressure between the gear wheels, as well as the pressure or force transmitted to the center of gear *II* is, $P_1 = \frac{P}{\cos 23°} = \frac{2,518}{.9205}$ = 2,735 lb. This force P_1 is exerted at an angle of 23° with respect to a perpendicular line. The reaction of this force or pressure of 2,735 lb. produces a like pressure on gear *2*, but in the reverse direction, upwards, as noted. Considering the mainshaft first, which carries gear *II*, (the countershaft bearings will be considered later) the pressure of 2,735 lb. is supported by bearings H_3 and H_4, (the latter is the roller bearings in the main drive pinion *4*). Figure 118 shows this layout diagramatically.

By the principle of moments, and taking moments about H_3, we have $6\frac{1}{2} \times 2,735 = 13\frac{1}{4} \times H_4$, and $H_4 = \frac{6.5 \times 2,735}{13.25}$ = 1,342 lb.

Taking moments about H_4, we have 6.75 × 2,735, = 13.25 × H_3, hence $H_3 = \frac{6.75 \times 2,735}{13.25}$ = 1,393 lb. Thus an additional pressure of 1,342 lb. is exerted on pinion *4*, at bearing H_4, this pressure being $1\frac{3}{4}$ in. from bearing H_2, and $4\frac{3}{4}$ in., from bearing H_1.

Taking moments about H_1 we find the pressure on $H_2 = \frac{4.75 \times 1,342}{3}$ = 2,125 lb. and on $H_1 = \frac{1.75 \times 1,342}{3}$ = 783 lb.

(Students are advised to familiarize themselves thoroughly with the subject of moments by reading Chap. II, pages 16 to 30.)

The next step is to consider the directions in which these pressures are exerted on the bearings, and for this purpose the gear *4* has been added, in dotted lines, (to Fig. 117), to show it on one common center with gear *II*, as is the case in practice.

Add the direction of the force of 1,600 lb. of gear *4*, Fig. 115, to that of Fig. 117, as shown in dotted lines. It can now be perceived that the two pressures on each bearing do not act in the same direction, nor entirely opposite to each other, but they form an angle of $180° - (23° + 23°) = 134°$ between each other.

Examining bearing H_2 (Fig. 119) we find 2,400 lb. (as calculated before) due to gear *4* driving gear *5*, and 2,125 lb. due to *2* driving the gear *II*. The direction of these forces on the gear wheels can be read off from Fig. 117, and it is noted that they form an angle of 134° between them. The resultant pressure can be found by the parallelogram of forces. That is to say draw a line c to some convenient scale, for instance 1 in. = 100 lb. then this line will be 24 in. long. Next lay off line e, at an angle of 134°, drawn to the same scale, thus 21.25 in. long. Now complete the parallelogram by drawing lines c_1 and e_1 parallel to c and e and of the same lengths. The resultant R is now drawn as shown, and by measuring its length, which is 16.35 in., the amount of the bearing pressure can be read off directly as equalling 1,635 lb. The resultant can also be found with the aid of trigonometry by the formula $R = \sqrt{c^2 + e^2 + 2ce \cos 134°}$. The cos of 134° is the same as the cos of 46°, (180 − 134 = 46°), but it is negative; cos 46° = .6946; cos 134° = −.6946, hence

$$R = \sqrt{2{,}400^2 + 2{,}125^2 + (2 \times 2{,}400 \times 2{,}125 \times -.6946)}$$
$$= \sqrt{5{,}760{,}000 + 4{,}410{,}000 - 7{,}500{,}000}$$
$$= \sqrt{2{,}670{,}000} = 1{,}786 \text{ lb.}$$

The above can be calculated in a few minutes with the slide rule or with the aid of logarithms. In bearing H_1 we had a pressure of 800 lb. due to the load on driving gear *4*, and 783 lb. due to load on driven gear *II*, and these two pressures form an angle of 134° between them.

Hence the resultant pressure is:

$$R = \sqrt{800^2 + 783^2 + (2 \times 800 \times 783 \times \cos 134°)}$$
$$= \sqrt{640{,}000 + 613{,}000 - 861{,}000}$$
$$= \sqrt{392{,}000} = 625 \text{ lb. approximately.}$$

Considering now the bearing pressures of the countershaft, (Fig. 126) (see also Fig. 112) we have seen that pressure P_1 of gear *5* is 1,600 lb. and P_1 of gear *2*, 2,735 lb.
Hence bearing pressure in:

$$(H5, \text{ due to gear } 5) = \frac{13.875 \times 1{,}600}{15.25} = 1{,}456 \text{ lb.}$$

$$(H6, \text{ due to gear } 5) = \frac{1.375 \times 1{,}600}{15.25} = 144 \text{ lb.}$$

$$(H5, \text{ due to gear } 2) = \frac{6.875 \times 2{,}735}{15.25} = 1{,}233 \text{ lb.}$$

$$(H6, \text{ due to gear } 2) = \frac{8.375 \times 2{,}735}{15.25} = 1{,}502 \text{ lb.}$$

Combining the pressures 1,456 and 1,233 in bearing 5, at an angle of 134° (in a similar manner as was shown in Fig. 119) we obtain the resultant of 1,070 lb. In bearing 6 we find the resultant to be 1,406 lb. The results may be tabulated as follows:

Pressures in bearing	1	2	3	4	5	6
Low speed						
Second speed	625	1,786	1,393	1,342	1,070	1,460
Third speed						
High speed (direct)	Very low-bearing pressures in transmission					
Reverse						

In addition to the bearing pressures, the speed of rotation must be known, as the capacities of bearings change with the speed. In the second speed the ratio of the clutch gear shaft to the main shaft is 3.23 (see page 128), hence, if the motor speed is 1,500 r.p.m., the speed of bearings H_3 and H_4 is $\frac{1{,}500}{3.23} = 465$ r.p.m.

The engine speed is to the countershaft speed as 41 teeth are to 19, $\frac{41}{19} = 2.158$. The countershaft speed is therefore $\frac{1{,}500}{2.158} = 695$, and bearings H_5 and H_6 must be selected accordingly. H_1 and H_2 run at engine speed. It should be noted that bearing H_3 supports one end of the propeller shaft and is made in practice considerably stronger, about 180 per cent of the capacity found on the second speed and given in the table above.

In figuring the bearing pressures for the other speeds, it should be remembered that in this transmission the torque on the clutch gear shaft is always 2,800 lb.-inches (the torque of the engine) and that on the countershaft it is always 6,043 lb.-inches, except on the high speed, when the countershaft is running idle. The resultant pressure in the bearings will therefore only change as a result of sliding the first, second, third or reverse gears into mesh, and taking their pitch radii into account. The results of the various bearing pressures when figured out may be added to the table. Note that gear *I* when meshed with gear *1* is 1¼ in. from bearing *H3* (see Fig. 112), while gear *1* of the countershaft is 1½ in. from bearing *H6*.

It is important to consider the bearing pressure of the various bearings on the reverse speed, for these will vary considerably if the reverse idler is placed to the right or the left of, or above or below, the countershaft. (In most cars the countershaft is below the main shaft; in the Class B transmission, they are both situated in a horizontal line.) Suppose the torque in the counter-shaft is 6,000 lb.-inches and the reverse pinion (of the counter-shaft) has a pitch radius of 1.6 in. The reverse is obtained (in the Class B transmission) by sliding the reverse idler gears (see Fig. 126*b*) *6* into mesh with gear *2* of the countershaft (see Transmission assembly drawing) and *7* into mesh with *1* of the main shaft. It was shown before (Fig. 117) that the force P_1 acting on gear *2* of the countershaft, is 2,735 lb. and that the pressure on bearing *H5* due to this gear was 1,233 lb. and on bearing *H6*, 1,502 lb. When the reverse gear is thrown into mesh, P_1 will still be 2,735 lb. but it will not be at an angle of 134° with respect to the direction of the force of 1,600 lb. (which arises from clutch gear driving the countershaft as indicated on Fig. 117), hence the pressures or forces on the bearing will not form an angle 134° with each other. In the Class B transmission the reverse idlers

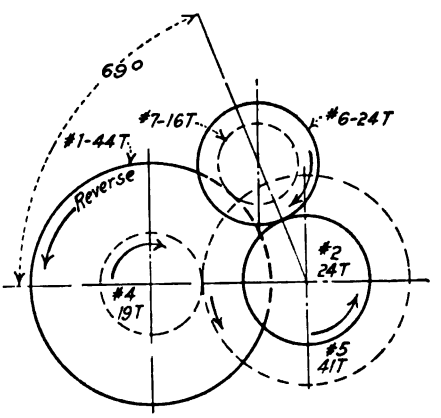


FIG. 121.

are on the top between the main shaft and the countershaft; the reverse idler shaft forming an angle of 69° with a line drawn through the axes of the main shaft and the countershaft, as shown in Fig. 121. (See also Figs. 115 and 117.) The gear *1* is now driven in the reverse direction due to the interposition of the idler gears *6* and *7*. Hence the load of 2,735 lb. (shown on Fig. 117) of the countershaft, will now be deflected 69° to the right, and the bearing load on the bearings *H5* and *H6* (arising from this 2,735 lb. force) will be deflected to the right. Figure 122 shows the forces on the countershaft due to the second speed forward. Figure 123 shows the forces due to the reverse. It must be remembered that gear *2* of the countershaft (the second speed gear) is used to drive the reverse idler *7*. Hence the bearing pressure on the countershaft arising from gear *2* (1,233 lb. on

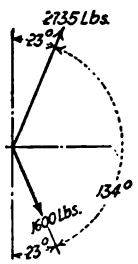


Fig. 122.

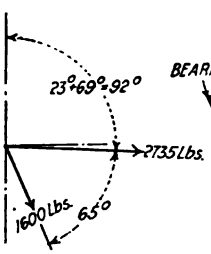


Fig. 123.

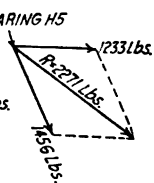


Fig. 124.

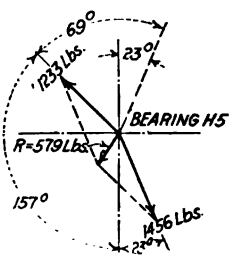


Fig. 125.

bearing *H5*, and 1,502 lb. on *H6*) will be deflected to the right 69°. Figure 124 shows the forces on bearing *5*. The force of 1,456 lb. arises from the clutch gear *4* driving the gear *5*. The resultant bearing pressure will now be 2,271 lb. instead of 1,070 (as shown for the second speed in the table on page 136) for $R = \sqrt{1{,}233^2 + 1{,}456^2 + 2 \times 1{,}233 \times 1{,}456 \times \cos 65°} = 2{,}271$ lb. If, as an example, the reverse idlers were placed at the bottom of the main shaft, instead of above, the bearing pressure will be entirely different, for in this case, the force or pressure of 1,233 lb. would be deflected 69° to the left, as indicated in Fig. 125. The resultant R is now = 579 lb., it is found by the same formula as before, but it must be noted that the cosine of 157° has to be considered, and this is the same as the cosine of 180 − 157 = cos of 23°, but its value is negative. Hence cos 157° = −.9205. In the same manner it will be found that the bearing pressures of the main shaft and that of the reverse idler

shaft is affected by the location of this idler shaft. Whenever the design permits and no other advantages are sacrificed, it is more economical to locate the shafts so that bearing pressures be as low as possible.

In transmissions, where the countershaft is placed below the main shaft, the pressures on the idler shaft bearings, (when the reverse is in mesh) are considerably lower when this idler shaft is located to the right instead of to the left of the countershaft, viewed from the front of the transmission.

Examples of Standard Transmission Designs.—The transmission of the Class B, U.S.A. military truck, has a nominal capacity of 3½ tons, but may be compared with 5-ton commercial models as far as strength is concerned. Figure 112 is a horizontal section showing that the transmission shaft *S* (often called the main shaft or primary shaft) and the countershaft *C* (often called the secondary shaft) are in a horizontal plane. Figure 126 shows a vertical section of this transmission assembly. The said shafts are placed in the box in a horizontal plane to obtain a ground clearance of 18 in. under the gear box (in most designs the shafts are underneath each other). Figures 126*a* and 126*b*, show the rear elevation and a plan view respectively of this gearset. The last figure discloses the idler gear shaft for obtaining the reverse speed; for reversing the rotation of shaft *S*, gear *6* is caused to mesh with gear *2* of the countershaft and *7* with gear *1* of the mainshaft. This transmission is placed amidships and is supported at three points; two forgings carrying lugs are bolted and doweled to *K* at the rear of the aluminum case, whereby it is supported by a frame cross member, while the front is supported by a trunnion *M*, 6 in. in diameter, which rests in a forging held by brackets riveted to the frame. The bearing H_1 is located inside of trunnion *M* (see Fig. 112). The weight of the case is 230 lb. (see description in the S.A.E. Transaction, 1918, Part I, by C. T. MEYERS, and A. M. COPELAND).

The breather *B* is shown enlarged in (Fig. 126*a*). The center line of the main shaft is somewhat lower than the engine shaft, necessitating the propeller shaft between engine and transmission to run at a slight angle. This lowers the gear box cover below the top of the frame, and reduces the angle of the rear propeller shaft connecting the gear box with the rear axle. Figures 127, 128 and 129 give detail drawings of the transmission case of the

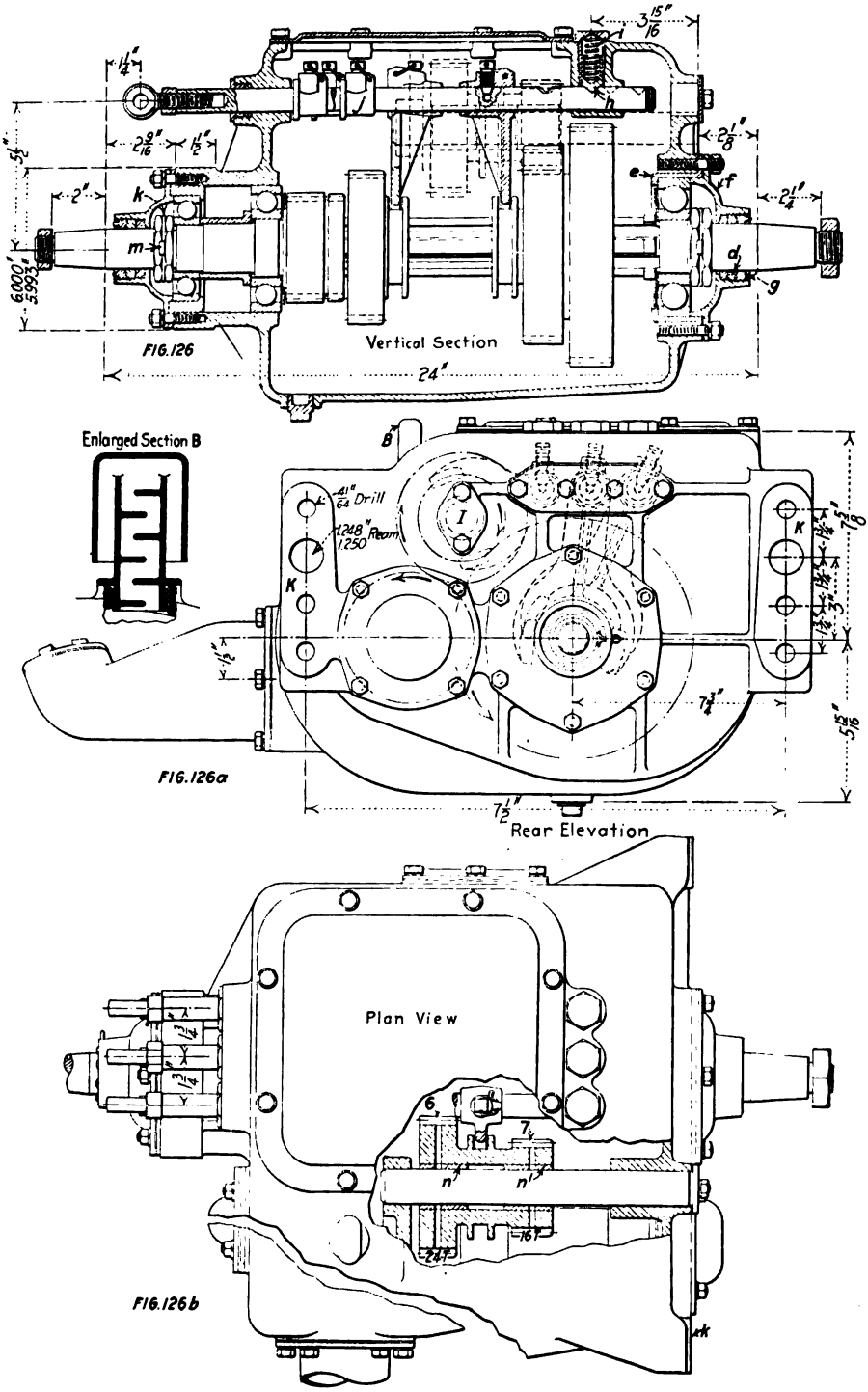


FIG. 126

FIG. 126a

FIG. 126b

FIGS. 126 to 126b.—Class B transmission.

Class B truck. Figure 127 shows a front elevation and a sectional view through the case. The thickness of the metal is ¼ in. except on the top of the case where it is 5/16 in., finished, to

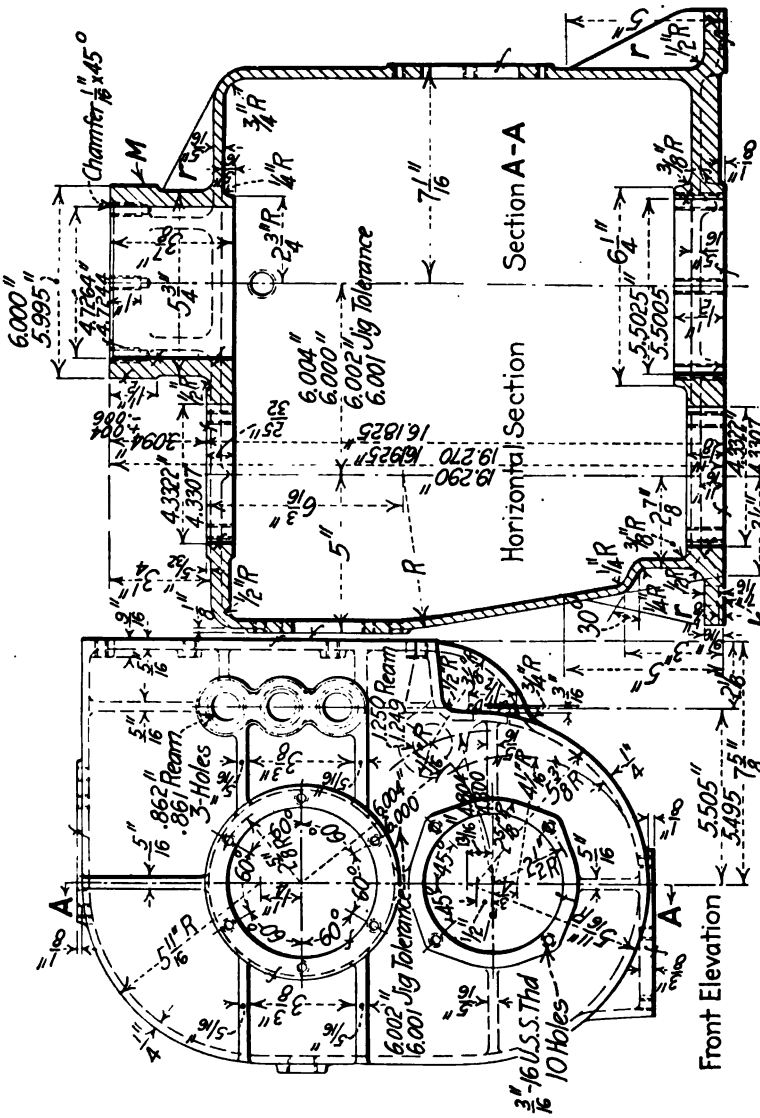


Fig. 127.—Class B transmission case.

receive the gear cover. It is also strengthened wherever required for retaining the bearings. At *K*, where the rear supports are attached, and at the trunnion *M*, by which the front of the case

is supported, the metal is $\frac{9}{16}$ in. thick, with ribs *r* giving additional strength to the supporting flanges.

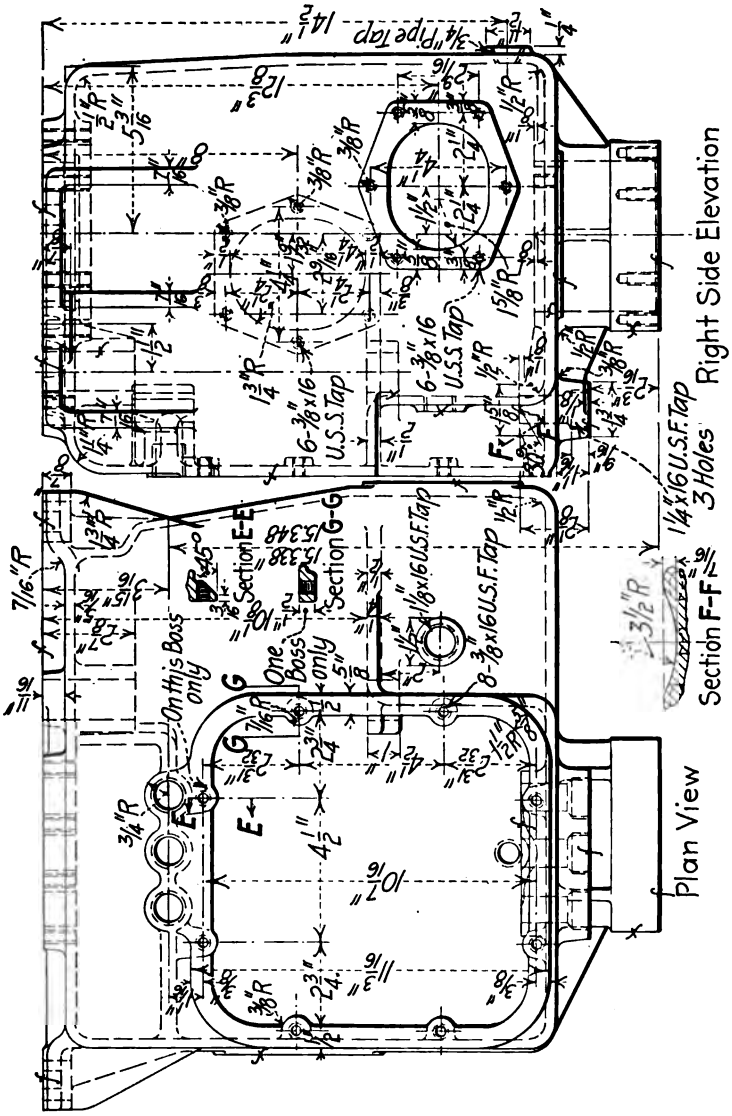


FIG. 128.—Class B transmission case.

Figure 128 shows a plan view and the right side elevation while Fig. 129 gives the rear elevation and some sectional views.

Figure 130 is the clutch gear (often called the main drive gear) and shank. As noted, it is made of S.A.E. 2320 nickel steel,

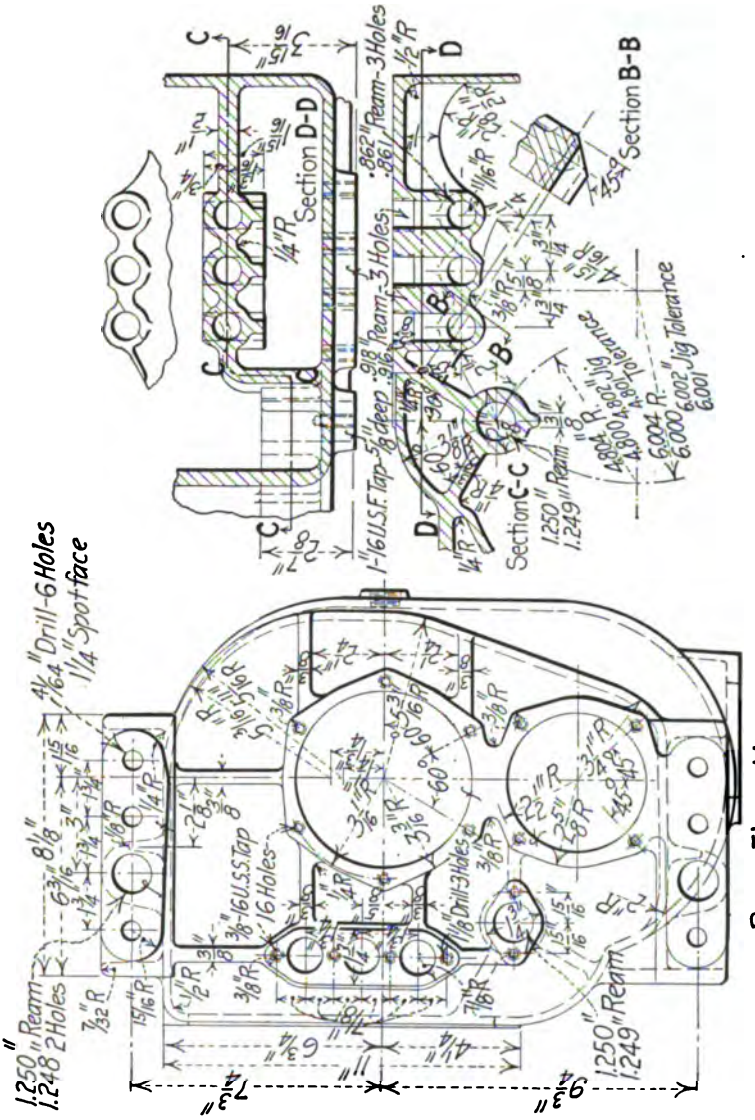


FIG. 129.—Class B transmission case.

the scleroscope hardness being from 80 to 90 on the gear, and on the rest of the shaft from 40 to 50. Being low in carbon contents,

this steel is carbonized in order to have the gear surfaces hard while the core remains soft. All the gears of this transmission are 20° involute 5-7 pitch. The part of the gear marked *a* is always in mesh with the countershaft gear. The front ends of the teeth, marked *b*, for a length of $\frac{5}{8}$ in. are rounded off, as seen in the right hand view, and are brought into mesh with teeth *IV* of gear wheel *3* (Fig. 112) thereby interlocking these two gears to form the direct drive for the fourth speed. The main drive gear has 19 teeth, and a pitch diameter of 3.8 in.; the end of its shank has a standard $1\frac{1}{2}$-in. S. A. E. taper, to which a

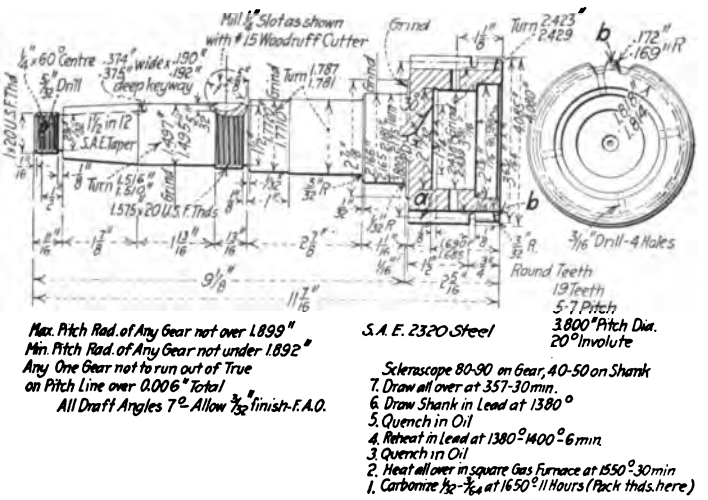

Mill ¼" Slot as shown with #15 Woodruff Cutter
Grind
Turn 2.423" 2.429"
.172" .169" R
¼" x 60° Centre
.374" wide x .190" .375" deep keyway .192"
⅝" Drill
1½ in 12 S.A.E. Taper
Turn 1.787 1.781
Grind
1.575 x 20 U.S.F. Thds
Turn 1.516" 1.512"
3/16" Drill-4 Holes
Round Teeth
19 Teeth
5-7 Pitch
3.800" Pitch Dia.
20° Involute
S.A.E. 2320 Steel
Max. Pitch Rad. of Any Gear not over 1.899"
Min. Pitch Rad. of Any Gear not under 1.892"
Any One Gear not to run out of True on Pitch Line over 0.006" Total
All Draft Angles 7° Allow 3/32" finish-F.A.O.
Scleroscope 80-90 on Gear, 40-50 on Shank
7. Draw all over at 357-30 min.
6. Draw Shank in Lead at 1380°
5. Quench in Oil
4. Reheat in Lead at 1380°-1400°-6 min.
3. Quench in Oil
2. Heat all over in square Gas Furnace at 1550°-30 min
1. Carbonize 1/32-3/64 at 1650° 11 Hours (Pack thds. here)

FIG. 130.—Clutch gear.

universal joint and shaft is attached connecting it with the clutch. Inside the gear a roller bearing is situated, which form the main shaft spigot or pilot bearing, the rollers running directly on the pilot *c* (Fig. 131) on the end of the mainshaft and inside the clutch gear. All the details are given on this drawing.

Figure 131 is the transmission shaft having standard S.A.E. six splines; the width of the splines is accurately ground, also the bottom of the splines on which the gears are sliding. In fact, the shaft is ground all over except where threaded, but a clearance is provided over the splines in the key-ways of the gears, the working fit being between the ground bore of the gears and the small diameter of the shaft between the splines.

The government left to the manufactures the choice of two materials, specifying either 3½ per cent nickel steel of low carbon contents and properly carbonized (which is the usual practice),

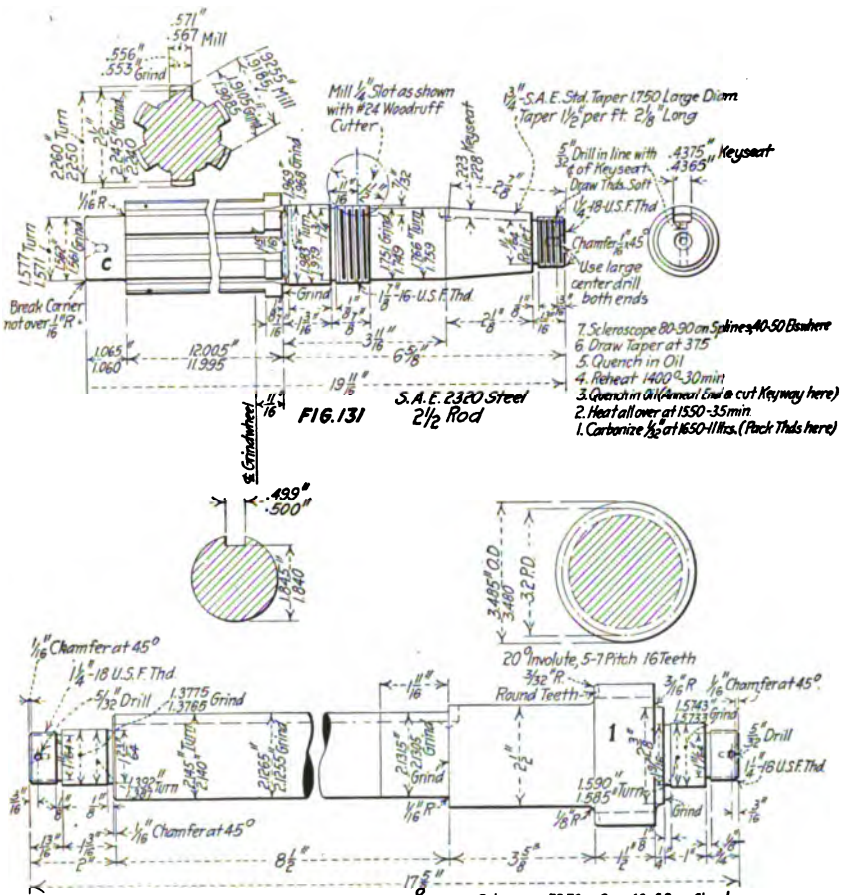

.571"
.567 Mill
.556"
.553" Grind
Mill ¼" Slot as shown with #24 Woodruff Cutter
¾" S.A.E. Std. Taper 1.750 Large Diam. Taper 1½" per ft. 2⅛" Long
.5" Drill in line with Keyseat
.4375" .4365" Keyseat
Draw Thds. Soft
¼-18 U.S.F. Thd
Chamfer 1/16" 45°
Use large center drill both ends
Break Corner not over 1/16" R
1/16" R
1/8"-16 U.S.F. Thd
Grind
7. Sclerascope 80-90 on Splines, 40-50 Elsewhere
6. Draw Taper at 375°
5. Quench in Oil
4. Reheat 1400°-30 min.
3. Quench in Oil (Anneal End & cut Keyway here)
2. Heat all over at 1550-35 min.
1. Carbonize 1/32" at 1650-11 Hrs. (Pack Thds here)
S.A.E. 2320 Steel
2½ Rod
FIG. 131
¼ Grindwheel
.499"
.500"
3.495" O.D.
3.460"
3.2 P.D.
20° Involute, 5-7 Pitch, 16 Teeth
3/32" R. Round Teeth
3/16" R
1/16" Chamfer at 45°
¼-18 U.S.F. Thd
5/32" Drill
1.3775
1.3765 Grind
5/16" Drill
½"-16 U.S.F. Thd
1/16" R
1/8" R
Grind
All Draft Angles 7°
Allow 3/32" finish
F. A. O.
Max. Pitch Rad of Any Gear not over 1.598
Min. Pitch Rad of Any Gear not under 1.592
Any One Gear not to run out of True on Pitch Line over .005 Total
Sclerascope 72-78 on Gear, 40-50 on Shank
7. Anneal Threads in Lead
6. Draw all over at 375°-30 min.
5. Quench in Oil
4. Reheat at 1380-1400
3. Quench in Oil
2. Heat at 1550, 15 min.
1. Carbonize 1/32"-3/64" at 1650, 11 Hrs.
FIG. 132

FIG. 131—Transmission shaft. FIG. 132.—Countershaft (Class B truck).

or the same metal with high carbon contents properly heat treated. The threads are drawn soft and so marked on the drawing, for if threads are hard they are liable to break off. This shaft is made

of 2½-in. round stock, the dimension over the splines being, turned 2¼ in. and ground 2.240 to 2.245 in.

Figure 132, is the countershaft which is forged integral with the low-speed gear. This shaft is provided with one key-way for a long key for a driving fit of the drive gears of the second and third speeds and the permanently meshed countershaft gear *5*. From .015 to .020. in. is allowed for grinding of the shaft.

Figure 133 is the countershaft drive gear which is permanently in mesh with the main drive gear. It has 41 teeth and a pitch diameter of 8.2. All the gears are made of 2320 S.A.E. nickel steel, carbonized, S.A.E. heat treatment *G*. The gears are forged, a draft angle of 7° being allowed.

Figure 134 is the third speed gear of the countershaft, having a pitch diameter of 6.6 (33 teeth). This and the preceding gear have a working face width of 1⅜ in. while the lower gears and the reverse have a 1½ in. width, since the torque is higher at the lower gears.

Figure 135 is the third and fourth speed gear, sliding on the transmission shaft. Note the 19 holes drilled to form the internal teeth to mesh with the clutch gear. This gear is shifted to the right by means of the main shift fork (Fig. 143), to produce the third speed, and to the left to lock it with the clutch gear for the fourth or direct speed.

Figure 136 is the first and second speed sliding gear hub to which the first and second speed gears are riveted as seen in Fig. 112. This hub is made of carbon steel S.A.E. 1020, and is carbonized. The number of teeth of the other gears not shown here may also be seen from Fig. 112, and when the pitch of the gear is known the pitch diameter can easily be found as described under gears. Figure 137 gives details of the nickel-steel key of the countershaft which is made from .0005 in. to .0015 in. wider than the slot in the gears, for a drive fit.

Figure 138 is the countershaft bearing adapter made of malleable iron, a cork gasket 1/16 in. thick is placed between this adapter and the countershaft retainer, (Fig. 139), thereby making the gearcase oil tight at the countershaft bearings. The latter is a steel stamping made of hot rolled stock, 3/16 in. thick.

The mainshaft bearing adapter *e* (see Fig. 126) is held in place by the mainshaft retainer (Fig. 140) which is studded to the case, as shown. A gasket is provided between *e* and *f*, also a packing on the inside, as seen at *d*. Figure 141 is the mainshaft

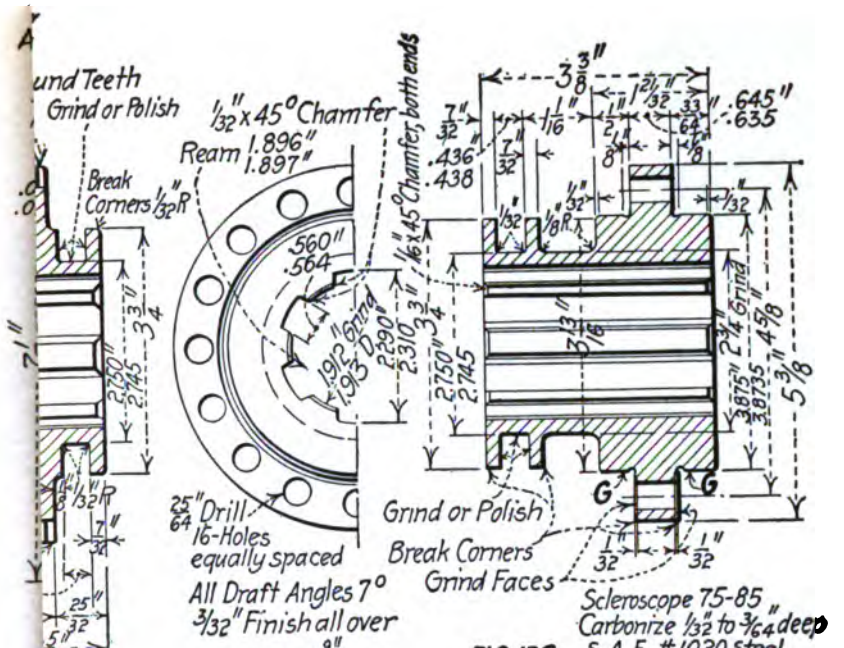


FIG. 136

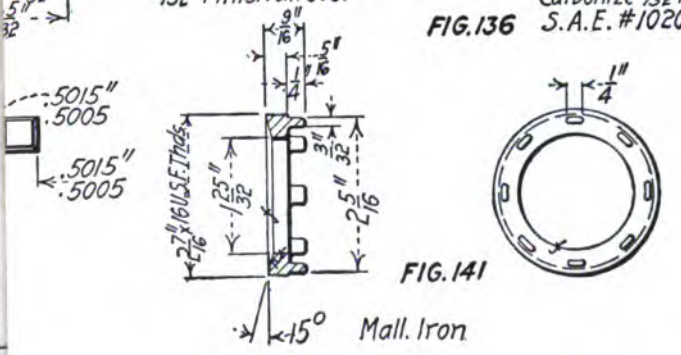


FIG. 141

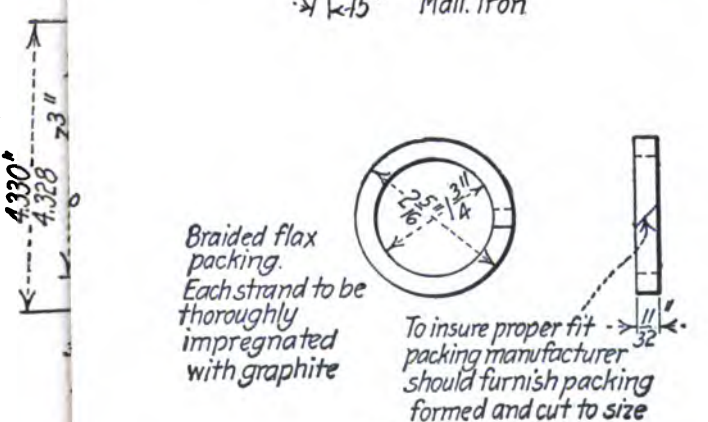

4.330"
4.328"
Braided flax packing. Each strand to be thoroughly impregnated with graphite
To insure proper fit packing manufacturer should furnish packing formed and cut to size

FIG. 142

FIG. 13 ountershaft key. FIG. 138.—Bearing adapter. FIGS. 139 and 140.—Retainer.

(Facing Page 146)

packing gland shown at *g*, for tightening the braided flax packing, (Fig. 142.)

Figure 143 the main shift fork, is made of S.A.E. 1025 or 1035 steel, heat-treated. Note how the section is U shaped for strength. Figure 144 shows one of the shift rods seen in Fig. 126, to which the shifter forks are attached; the mesh locking plungers *h* (sometimes called spring plungers or poppets) may also be seen, forced into the notches of the shift rod by springs *i*, to hold the gears by means of the shift rods in their proper positions. The lower end of the lock plunger is wedge shape, machined

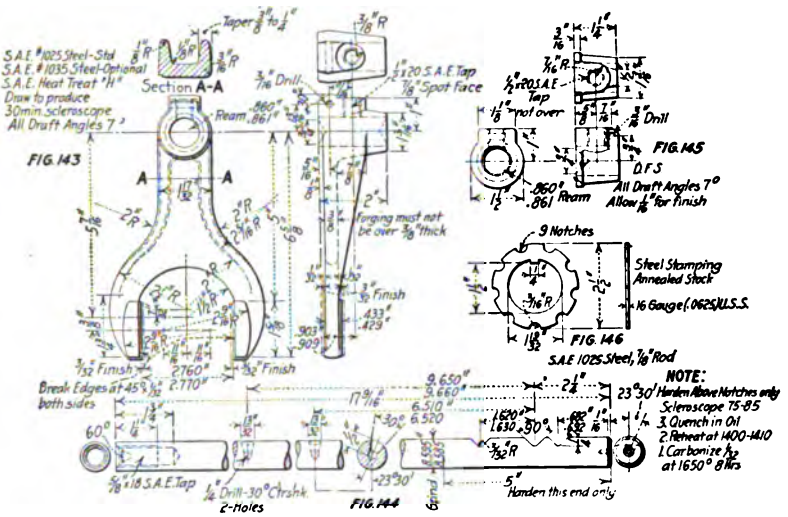


FIG. 143.—Shift fork. FIG. 144.—Shift rod. FIG. 145.—Shift rod stop. FIG. 146.—Clutch gear lock washer (Class B, transmission).

concave and rounded off, to give greater wearing surface than a simple cone pointed plunger, while the concave side prevents the plunger from turning around.

Figure 145 is a detail of the shift rod stop, (marked *j*, in Fig. 126), to prevent the gears from being shifted further forward than required. Figure 146 is the clutch gear lock washer shown at *k* in the assemblies. It is placed between the two nuts and after these are tightened, the ends of the lock washer are hammered over in one place to lock the outside nut and at another place for the inside nut, see *m* (Fig. 126). Annular ball bearings are used for the mainshaft and countershaft with the exception of the one roller bearing in the clutch gear for the pilot of the mainshaft;

plain phosphor bronze bearings are employed for the idler gears marked *n*, (Fig. 126*b*). It should be noted that the drive gear shank terminates in a 1½-in. S.A.E. taper, and the mainshaft in a 1¾-in. taper. The reason for the larger taper of the mainshaft is the increased torque produced at the lower speeds. The speed change H-plate is the S.A.E. standard. No rocker-arms are used in the transmission but the shift forks are permanently fixed to the rods by means of hardened screws having special taper ends, countersunk into the rods, and wired to prevent their coming off. The shift rod nearest the idler gear is used for

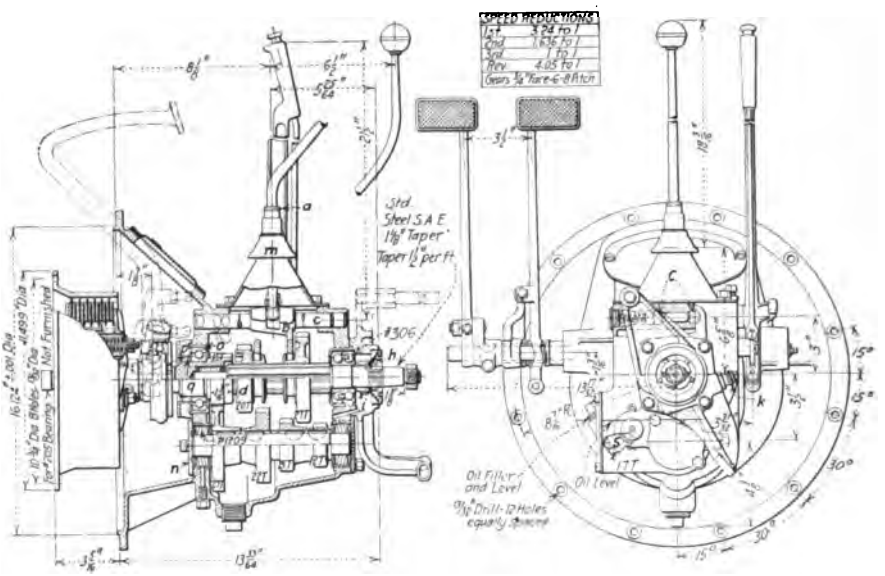


FIG. 147.—Transmission of Warner Gear Co.

the reverse. The reverse idler gears are a single forging with a shifter slot between the gears. All the gears being forged, hence a draft angle of 7° is specified on each gear drawing.

Figure 147 is the passenger car transmission manufactured by the Warner Gear Co. for cars weighing approximately 3,300 or 3,400 lb. The design of this transmission follows more or less conventional present day practice. It has a one piece unit power plant type of housing, made of either aluminum or cast iron as desired. This housing is provided with an opening over the idler gear corresponding to the small S.A.E. standard tire pump mounting.

The shift rails are carried inside the case as this has been found desirable for manufacturing reasons. The control pedestal is directly over the center of the gear set which brings it well forward in the car, close to the toe-board, allowing room between this pedestal and the heel-board for passengers to change from one side to the other on the front seat. The control and brake levers are bent back to the desired amount to bring them within easy reach of the driver. The control assembly is of the ball and socket type, and is held together by the knurled nut *a*, just above the cap, threaded on to the shank of the control lever.

The shift forks *b* (see also Fig. 148) are fastened to the shift rails by tapered screws, shown in (Fig. 149). This taper screw is very satisfactory from a manufacturing standpoint; it is also used to hold the throw-out yoke to the two throw-out yoke shafts of the clutch. The head of the screws are drilled and wired to prevent their coming loose. The gears are made according to common practice, from high carbon, 3½ per cent nickel steel stock and are oil-hardened (S.A.E. 2345). Figure 150 shows the high and intermediate sliding gear, marked *d* on the assembly drawing. Moving this gear to the left engages the ends of the teeth of the clutch gear (Fig. 151) which has 15 teeth, shaped as shown.

As seen from the assembly drawings the clutch gear shaft and the right side of the main shaft run on ball bearings while the countershaft and the pilot bearing of the main shaft are supported by Hyatt roller bearings. If Hyatt bearings were used on the main shaft it would be necessary to provide bronze and steel end washers. The annular bearing on the clutch gear is carried in an adapter *o*, clamped to the clutch gear journal by a snap-ring in a groove of the journal, thus making this bearing take all end thrust, which may arise on the clutch gear, in both directions. Similarily, the double ball bearing at the rear of the main shaft is held by its adapter to take the end-thrust in both directions. By making these two shafts take end thrusts independently of each other, it is claimed that any variation that may be due to machining does not cause any binding action between the end of the main shaft and the clutch gear. Note the 3/16-in. oil-hole through said clutch gear for lubricating the pilot bearing.

It might be mentioned that when the clutch is released, the thrust of the springs is communicated back, through the shank of the clutch gear, onto the clutch gear bearing, and this makes it

all the more necessary to retain this bearing as described above and it prevents the thrust of the springs from being communicated back to the main shaft.

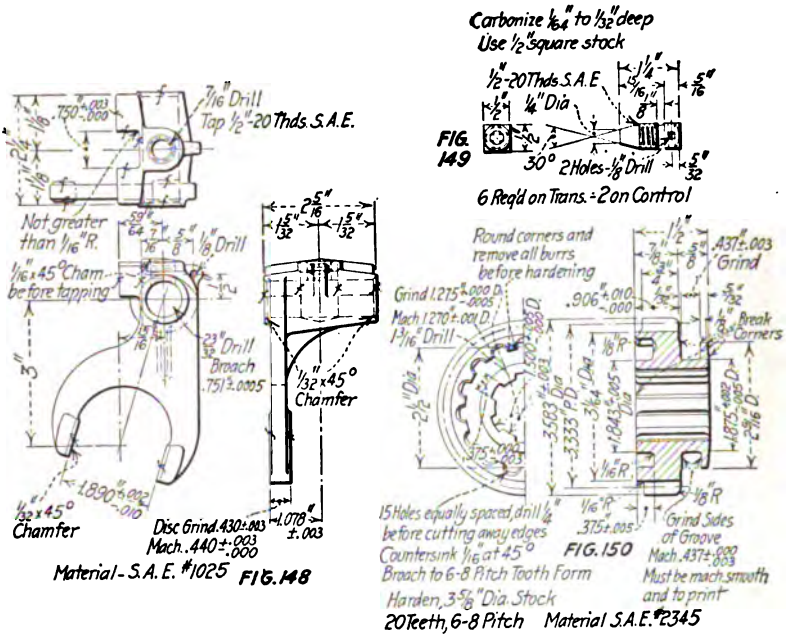


FIG. 148.—Shift fork. FIG. 149.—Tapered screw. FIG. 150.—High and second gear. FIG. 151.—Clutch gear. (*Warner Gear Co. Transmission.*)

No thrust washers are provided on the countershaft but the ends of the shaft are ground and bear directly against the inner

faces of the countershaft end plates. These plates are ground on the inner surfaces and are provided with oil-grooves to insure lubrication. Many concerns strongly recommend the use of oil, and not grease, for lubricating transmissions as the latter is too heavy to penetrate to certain surfaces such as the idler bushing. When grease is used as a lubricant, the gears, when rotating, tend to cut a path through the grease and throw it off to the sides of the case, thus leaving the gears at times practically without lubrication. In cold weather when the grease becomes very stiff gear-shifting is rendered difficult. In addition, oils are a much better lubricant than grease. In the gearset shown in Fig. 147 an oil-filler is provided at the side of the case in the form of a spout, which also acts as a level gauge when filling the gear box; when the oil reaches the level of the spout it will run out. In this transmission case, two quarts of oil are necessary to reach the height as shown, that is to say, it will cover the countershaft but not the countershaft gears.

The reverse idler gear (Fig. 152) is carried on a floating bushing (Fig. 153); the bushing is free both in the gear and on the shaft. A bushing of this kind is less likely to stick and it is more satisfactory from the standpoint of replacement than when the bushing is pressed into the idler gear.

The rear main shaft bearing retainer consists of either a plain retainer and dust-cap, or it may include the enclosed type of speedometer drive; in addition it may have arms as shown for supporting a transmission brake. The small gear *i* on the main shaft is used for driving the speedometer drive gear shaft *k* to which the speedometer flexible cable is attached. The rear of the main shaft is provided with an oil flinger *h* which is self explanatory. It is claimed that this is more satisfactory than a felt washer as there is nothing needing adjustment or wearing out, and the oil is retained perfectly. At the same time it acts like a breather opening for the transmission, thereby relieving any pressure that may arise through heat which would cause an expansion of the air. The control set is also opened at the top, under the skirt of the cap *m*, for the same purpose. If no breather hole is provided there is a tendency for oil leaks.

The spring plungers or poppets (Fig. 154), used on the shift rail, have their ends shaped like a wedge to present a larger surface for engaging the notches in the shift rail. (In many designs the plungers are simply cone shaped.) Note the $^{15}/_{16}$-in. radius

milled across the point of the wedge at the bottom, which causes the plunger to straddle the rail when out of its notch, thus preventing the plunger from turning around on its seat; it is similar to the plunger of the class B transmission. No other guide is found necessary to keep the plungers properly lined up.

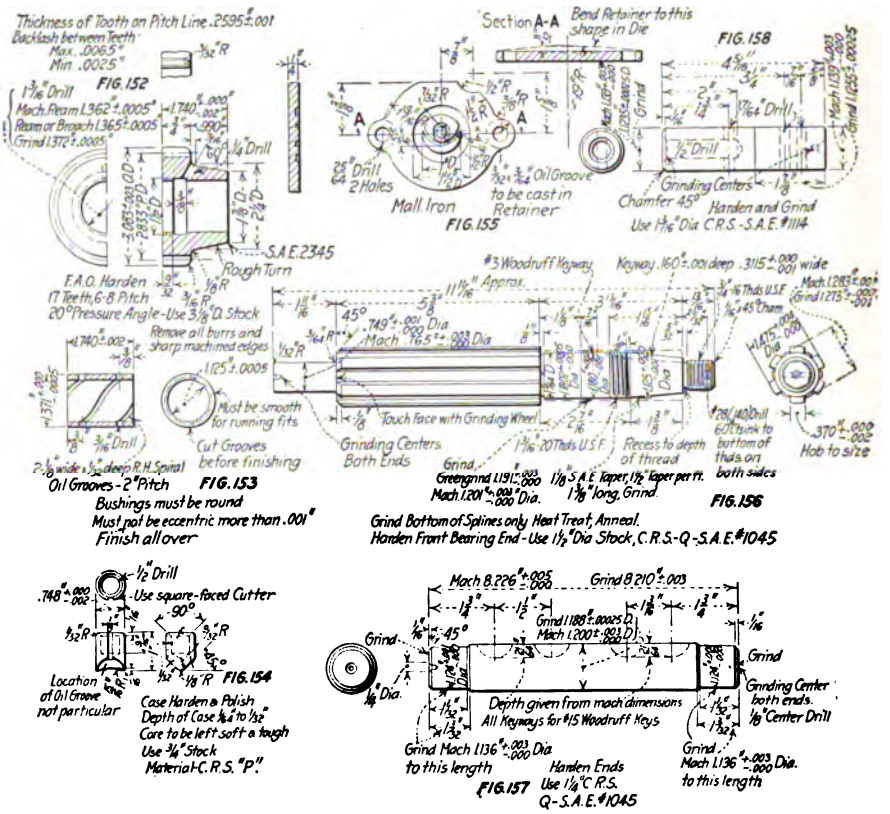


FIG. 152
FIG. 153
Bushings must be round
Must not be eccentric more than .001"
Finish all over
FIG. 154
Case Harden & Polish
Depth of Case 1/64 to 1/32"
Core to be left soft & tough
Use 3/4" Stock
Material-C.R.S. "P".
FIG. 155
Mall Iron
Oil Groove to be cast in Retainer
Bend Retainer to this shape in Die
Section A-A
FIG. 156
Grind Bottom of Splines only Heat Treat, Anneal.
Harden Front Bearing End - Use 1 1/2" Dia Stock, C.R.S.-Q-S.A.E. #1045
FIG. 157
Harden Ends
Use 1 1/4" C.R.S. Q-S.A.E. #1045
FIG. 158
Grinding Centers
Chamfer 45°
Harden and Grind
Use 1 5/16" Dia C.R.S.-S.A.E. #1114
Hob to size

FIG. 152.—Reverse idler gear. FIG. 153.—Bushing. FIG. 154.—Spring plunger. FIG. 155.—Bearing retainer. FIG. 156.—Transmission shaft. FIG. 157.—Countershaft. FIG. 158.—Idler gear shaft. (*Warner Gear Co. Transmission.*)

Figure 155 is the bearing retainer shown at n in the assembly. Note the oil grooves in this part, also in the bearing retainer p of the clutch gear shaft where it rides on collar q. Figure 156 is the transmission shaft, made of S.A.E. 1045 steel. All the dimensions are given and its careful study is recommended to students. Figure 157 is the countershaft. Figure 158 is the idler

gear shaft on which the gear (Fig. 152) and bushing (Fig. 153) rotate. This gear is shown at *S* in the right-hand view of the assembly.

Figure 159 *a*, *b*, and *c* show the transmission and clutch of the well-known Mack 1½- to 2-ton truck, manufactured by Mack Motors, Inc., formerly the International Motor Co.

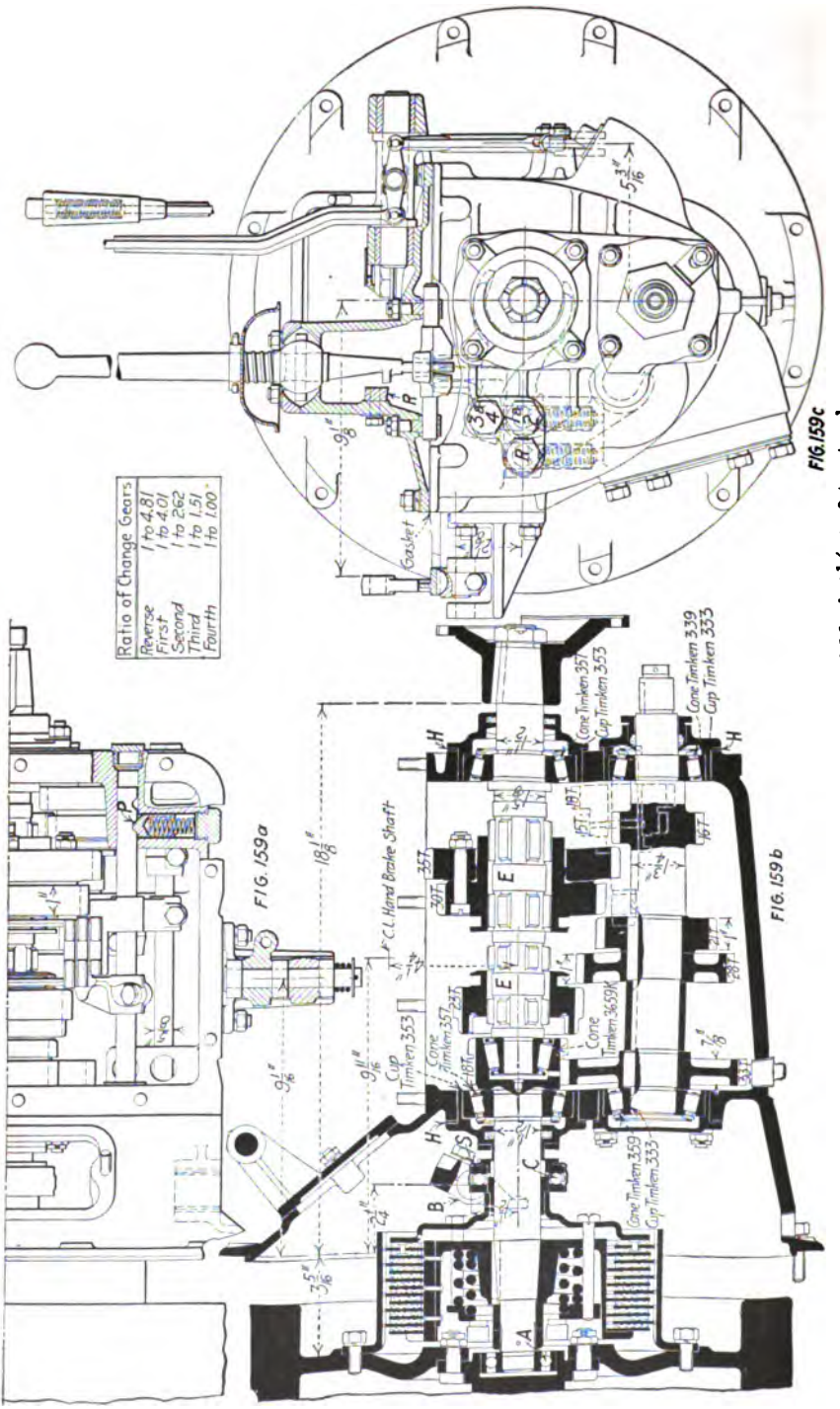
All the bearings of this transmission are Timken rollers, including that of the transmission shaft pilot. The pilot end of the main drive gear shaft, or clutch gear shaft, is provided with a ball bearing inside the end of the crankshaft as shown at *A*. *B* is the clutch release fork which bears against a thrust ball bearing *C*. The drawing shows the transmission as redesigned quite recently with the following changes as compared to the gearset they manufactured heretofore:—The clutch throw-out bearing is positively oiled, seepage from the front transmission bearing being led through a hole in the shaft to the bearing and centrifugally forced into it; it is retained in it by means of a stamping *S* which also makes a self-contained unit of the bearing.

The spigot or pilot bearing at the front of the spline shaft has been increased in capacity by the use of longer rollers. The countershaft gears are pressed and keyed on, while formerly they were riveted. It is claimed that this insures a higher degree of concentricity and alignment with permanent tightness of the shaft. The double sliding gears, those on the spline shaft for the first and second speeds, are bolted together instead of being riveted, for the same reasons.

The reverse idler shaft is positively locked against turning. Adjustments of bearings is made by adding or removing thin shims *H*, which, once set, cannot alter. The points *P* of the shifter shaft locking plungers have been rounded off so as to reduce wear on the shafts and the notch edges.

Instead of cap screws in the aluminum case, permanent studs are used for securing the various appendages. The case is made stronger and more clearance is allowed over the gears. It is fitted with an opening for attaching a tire pump or supplementary power take-off (the ordinary power take-off being on the extension of the countershaft). There is another orifice provided for establishing the correct oil level (by overflow) when filling.

An improved cover has been fitted, embodying an improved gear-selecting and shifting mechanism. It is self-locking, that is, automatically preventing the engagement of more than one



FIGS. 159a, b and c.—Transmission of Mack 1½- to 2-ton truck.

gear at a time, or the engagement of the reverse, except by intention. The hand lever is absolutely plain with no latch or finger-grip. Selecting reverse is accomplished simply by pressing *down* on the hand lever while shifting; this clears the catch *R*. On leaving the reverse, the lever automatically springs up into its normal position.

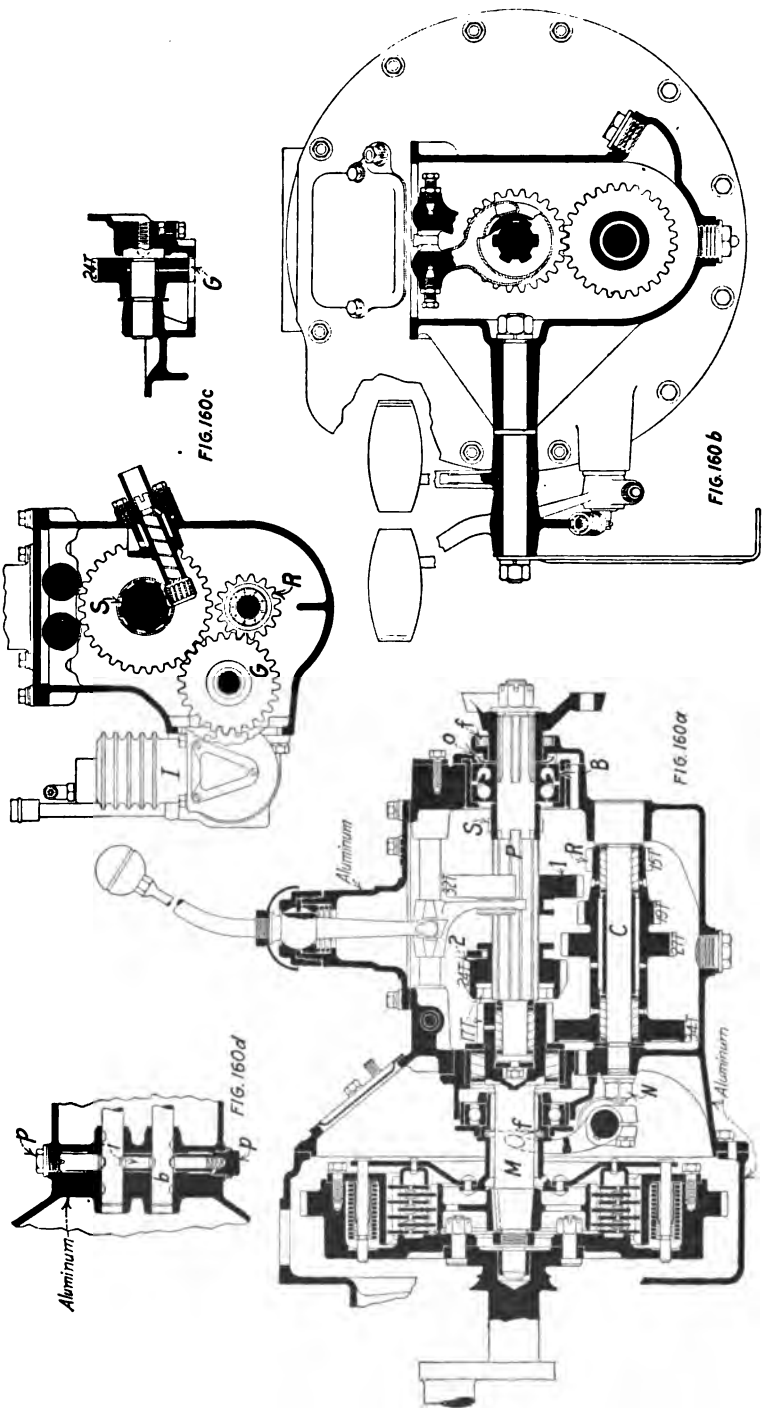
The spline shaft is of an improved type, known as the interrupted spline; the splines, instead of being continuous from end to end of the shaft, are cut away at intervals to leave short lands *E*, or plain cylindrical lengths, of a diameter ground to a precise fit with the inner diameter of the broached holes in the sliding gear bubs. The lands on the shaft are so disposed as to support both ends of the gear hubs when the gears are meshed; it is claimed that by these means, play is minimized and concentricity and parallelism of gear tooth elements to shaft centerlines is obtained. An excellent alignment decreases and more evenly distributes the wear on the gear teeth and is conducive to greater silence.

Self-lubricating bearings carry the brake lever shaft, supplanting the hand-oiled bushings previously fitted. Improvements of a minor nature have been made to the heads of the clutch adjusting screws; they are slotted to show the location of the concealed keys which lock them.

Figures 160*a*, *b*, and *c*, give three views of the transmission of the latest Packard six-cylinder car. This car has an L-head engine with 3⅜- by 4½-in. cylinders, cast in one block. The chassis weight is 2,250 lb.; with a touring car body about 3,100 lb. The gear ratios of the transmission are 3.368 to 1 on the first speed, 1.774 to 1 on the second, direct on the third, and 4.26 to 1 on the reverse. The gears are made of chrome-nickel steel.

As may be seen from the drawing, when the gear 1 is moved to the right it engages the reverse gear *G* (Fig. 160*c*) which is in constant mesh with gear *R*; moving it to the left establishes the first speed forward. This gearset is of the conventional type, equipped with ball bearings for the clutch gear shaft *M* and the transmission shaft *P*, while Hyatt roller bearings are provided for the countershaft gears, and the pilot of the main shaft.

All the countershaft gears are made integral, the two roller bearings at each end being seated on the fixed countershaft *C*, which is inserted from the right end of the case and is held in place by nut *N*.



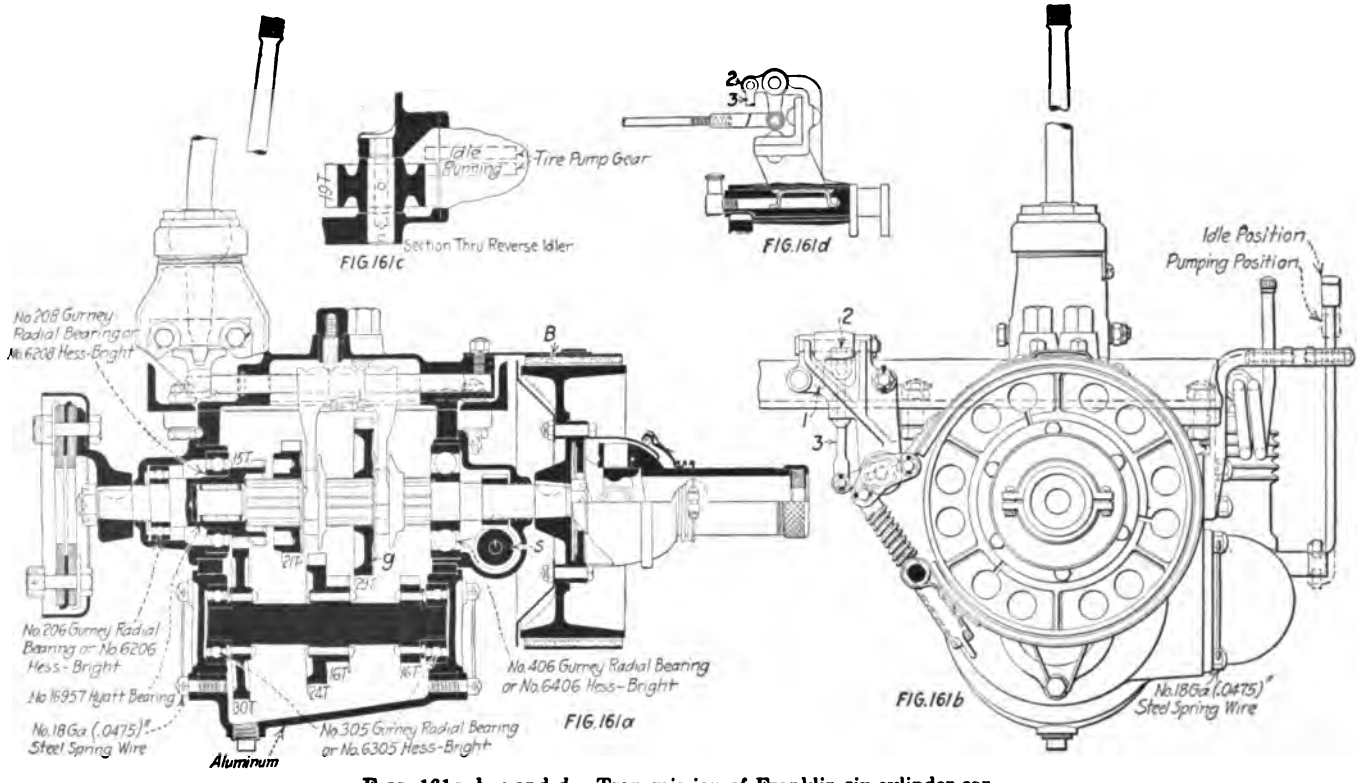
FIGS. 160*a*, *b*, *c* and *d*.—Transmission of Packard six-cylinder car.

The main bearing *B* of the mainshaft, which sustains the highest load due to the proximity of the reverse gears, and which also carries the forward end of the propeller shaft, consists of a double row New Departure ball bearing which takes end thrust in both directions.

Note the oilslinger *o* at the end of the mainshaft, in addition to the felt packing *f*, to prevent oil leaks. *S* is the speedometer gear, a section of which is seen in (Fig. 160*c*). The gear *G* is also used for driving the air pump situated at *I*. The constant mesh gears are allowed a back lash or play of from .004 to .008 in. Figure 160*d* shows the shifting bars *b*, held in their respective position by spring plungers fitted within hollow screws *p*. In addition there is a bar lock *i* which positively holds the second shifting bar against displacement until the first bar is returned to its neutral position; only in that case will the bar-lock release the second shifting bar.

Figures 161*a*, *b*, *c*, and *d*, show the transmission of the Franklin Model 9*B*, six-cylinder ($3\frac{1}{4} \times 5\frac{1}{4}$-in.) car. The width of the gears is $\frac{5}{8}$ in., except that of the constantly meshed gears which is $\frac{1}{2}$ in. In this gear set the shifter forks are fixed to the shifter bars and the latter are moved longitudinally by the gear shift lever to establish the various speeds. The brake *B* is described under transmission brakes. The air pump is driven from the reverse idler gear as seen from Fig. 161*c*. The speedometer drive is indicated at *S*. This transmission is placed amidship, hence one or two additional universal joints are necessary; the angularity between the transmission and the clutch is slight, hence a flexible disk type universal joint is here used, while between the gear case and the rear axle a spicer joint is employed.

The clutch gear has a face width of $\frac{1}{2}$ in.; it is a drop forging and receives the first heat treatment when forged. After the second heat treatment it receives the finish cut and it is then carbonized $\frac{1}{64}$ in. deep. Backlash of .005 in. is allowed when assembled with its mating gear on the countershaft. All the gears are of 6–8 pitch, involute teeth, with a pressure angle of 20 degree. Figure 162 is a detail drawing of the high and intermediate sliding gear of the transmission shaft. As seen from the drawing, the end *e* of the teeth have a special shape and mesh with similar teeth in the clutch gear, for the purpose of sliding them more easily into engagement for the direct drive. The gear has .007 in. backlash with its mate when assembled.



Figs. 161*a*, *b*, *c* and *d*.—Transmission of Franklin six-cylinder car.

The transmission shaft or sliding gear shaft, has a diameter of 1.269 to 1.267 in., at the bottom of the splines. At the large end of the taper, where attached to the transmission or propeller shaft brake hub, it is 1.0627 to 1.0633 in. Figure 163 is the countershaft which is shown integral with the reverse gear, but the assembly indicates that this shaft is not now made integral.

From the standpoint of quantity production it is sometimes more economical not to forge the shaft integral (it also saves the

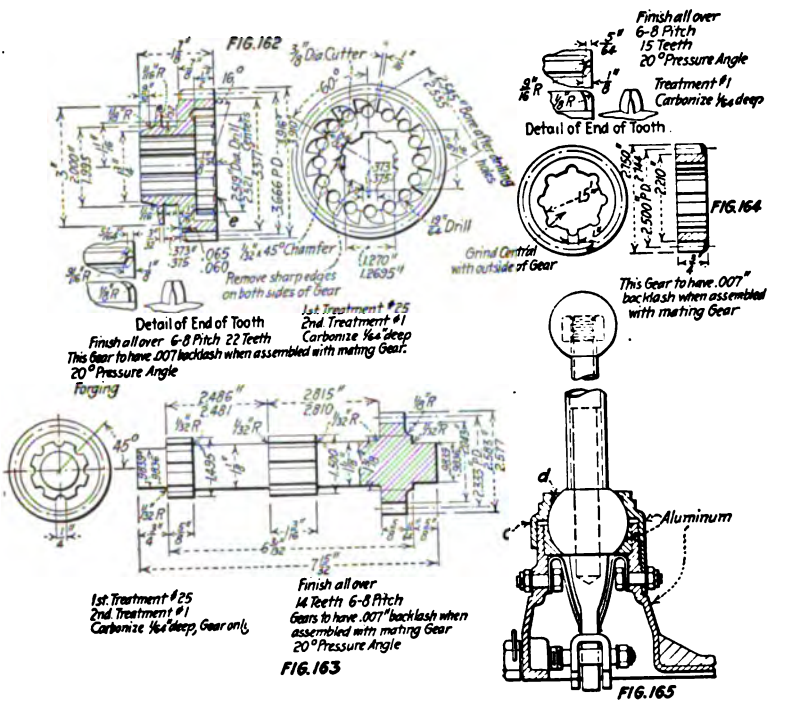


FIG. 162
Detail of End of Tooth
Finish all over 6-8 Pitch 22 Teeth
This Gear to have .007 backlash when assembled with mating Gear.
20° Pressure Angle
Forging
1st Treatment #25
2nd Treatment #1
Carbonize 1/64" deep
Remove sharp edges on both sides of Gear
Grind Central with outside of Gear
Drill
45° Chamfer
FIG. 163
Finish all over 14 Teeth 6-8 Pitch
Gears to have .007" backlash when assembled with mating Gear
20° Pressure Angle
1st Treatment #25, 2nd Treatment #1, Carbonize 1/64" deep, Gear only
Finish all over 6-8 Pitch 15 Teeth 20° Pressure Angle
Treatment #1 Carbonize 1/64 deep
Detail of End of Tooth
FIG. 164
This Gear to have .007" backlash when assembled with mating Gear
Aluminum
FIG. 165

FIG. 162.—Direct drive gear. FIG. 163.—Countershaft. FIG. 164.—Low-gear pinion. FIG. 165.—Gearshift lever and case cover (Franklin transmission).

cost in dies) but to turn it from round stock and then attach the reverse gear to the shaft in the same manner as is done with the other gears. The method of attachment can be seen from the assembly drawing, and from the detail, (Fig. 164), of the low gear pinion. When the gear is placed in position, the eight semi-circular rivet holes of the shaft correspond with similar holes of the gear, rivets are used to make a firm joint. The reverse idler gear is in constant mesh with the reverse pinion of

the countershaft. When the low speed gear *g* (see assembly) is moved to the right, it will engage the reverse idler, (not shown in the assembly but a sectional view of it is seen in Fig. 161*c*).

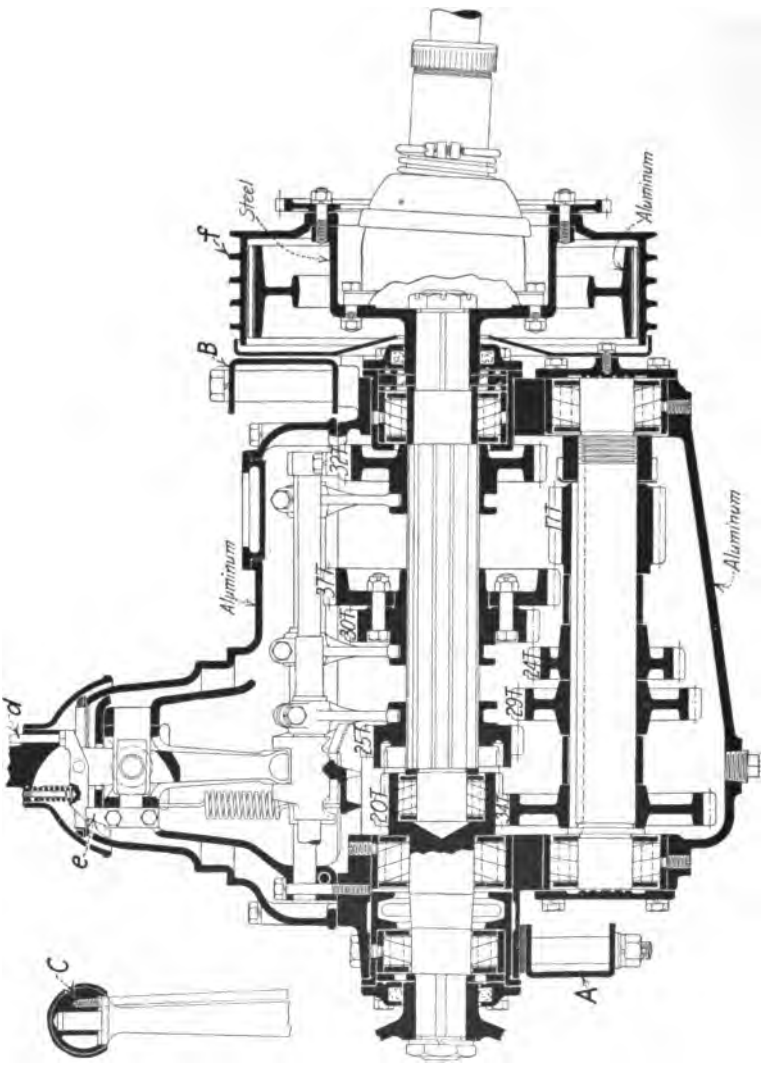


FIG. 166.—Transmission of Mercer car.

Figure 165 gives a section through the transmission case cover, disclosing the gearshift lever and the method of holding it in place by threaded cap *c*. A felt washer is placed at *d* to prevent

dust and dirt from reaching the ball and socket and to maintain lubrication.

Figure 166 is the transmission of the Mercer car. Hyatt roller bearings are used throughout and thrust washers are provided

FIG. 167.—Fuller transmission, unit powerplant type.

at the ends of the bearings as shown. This gearset has four speeds forward and reverse and it is suspended amidship, being connected with the motor by means of flexible couplings. The gears and shafts are made from chrome-nickel steel. A locking

device is provided which makes it impossible to shift gears unless the clutch is released, and unless the gears are in proper mesh the clutch cannot be re-engaged.

All the countershaft gears are keyed to the shaft by one key extending through all. The transmission shaft has six splines, but the ends of this shaft and of the clutch gear, to which the universal joints are attached, are square and are held in place by nuts as shown. The transmission case is supported at three points; in the front, at *A*, (in the center of the case,) and in the

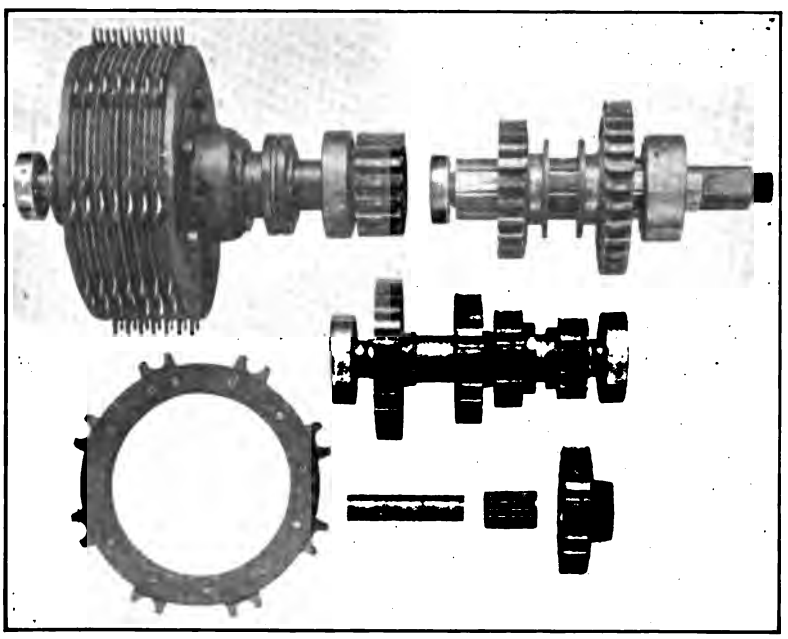


FIG. 168.—Clutch and transmission gears. (*Fuller & Sons Mfg. Co.*)

rear, at *B*, on two sides. *A* and *B* being channel cross members. To engage the reverse the handle *C* is pressed down, this forces rod *d* down and causes the small lever to clear the catch *e* which makes it possible to shift into the reverse.

Figure 167 is the transmission manufactured by the Fuller & Sons Mfg. Co. suitable for a motor of up to 227-cu. in. piston displacement, or for a ¾- to 1½-ton truck. It is of the unit power plant type, showing a tire pump mounted in position and a power take-off on the right, if desired to use power for other purposes. Figure 168 shows the clutch and the gears with the

shafts and their ball bearings. The single gear in the lowest right hand view is the "reverse" which runs on a roller bearing. This last transmission and clutch are used for a slightly larger

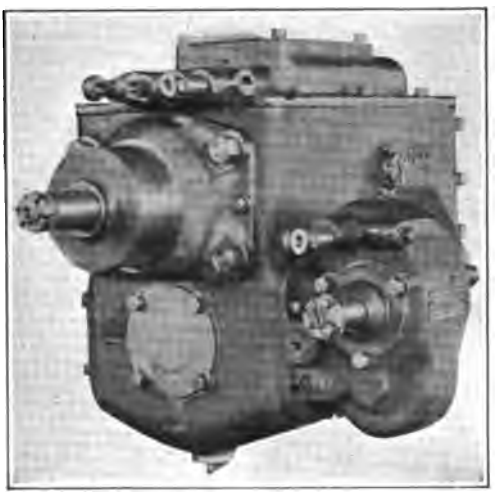


FIG. 169.—Fuller transmission, amidship type.

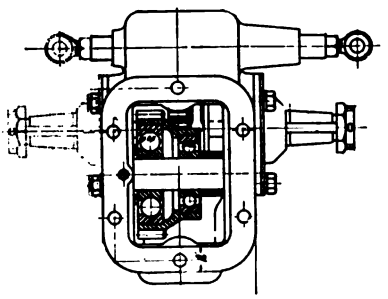
capacity motor (up to 294-cu. in. piston displacement) or for a truck of from 1- to 2-ton capacity.

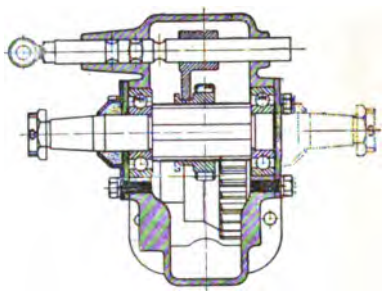
Figure 169 shows a transmission for amidship location,

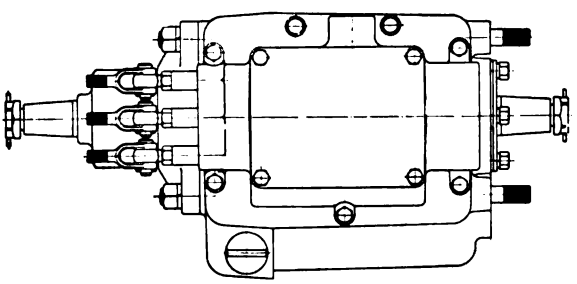
FIG. 170.—Power take-off attachment. (*Fuller & Sons Mfg. Co.*)

having four speeds forward and reverse, with gear ratios of 6.1 to 1, on the first speed; 3.2 to 1, on the second; 1.7 to 1, on the third; 1 to 1, on the fourth, and 6.3 to 1 on the reverse. The general

arrangement of this gearset is similar to that shown in Fig. 172. It is suitable for a motor of up to 450-cu. in. piston displacement, or for trucks having a capacity of from 3½- to 7-tons.







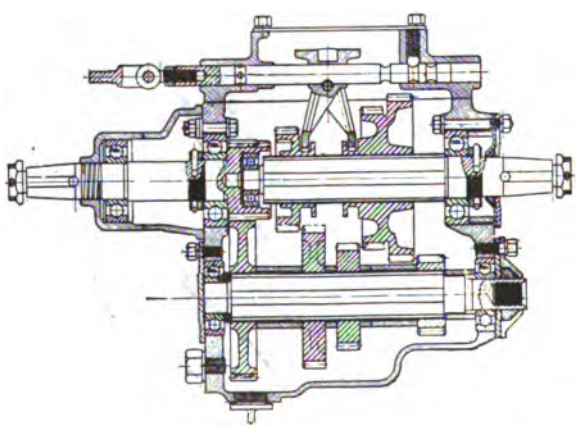


FIG. 171.—Details of power take-off attachment. FIG. 172.—Fuller transmission.

The clutch gear shaft has a 1½-in. S.A.E. standard taper, and the main shaft a 1¾-in. taper. The face width of the gears is 1¼ in., except that of the low and reverse gears which are 1½ in. wide.

Figure 170 illustrates the power take-off attachment, details of which may be seen from Fig. 171. The shaft which terminates into a 1/16-in. standard S.A.E. taper, may be reversed, end for end, (shown dotted), to change the take-off location. The gear of the shaft is shifted by the shift bar, shown in the upper right hand view, to engage the constant mesh countershaft gear, when the power take-off is to be used. The gear width (shown in Fig. 170) is 7/8 in. Figure 172 is somewhat smaller size of transmission

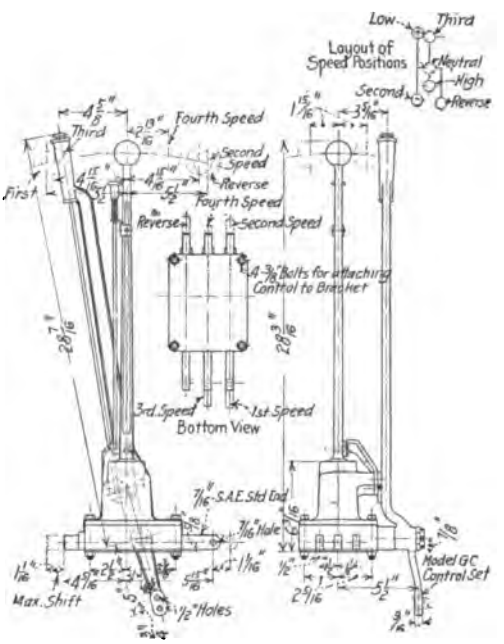


FIG. 173.—Fuller control set.

manufactured by the same company, suitable for a motor size up to 350-cu. in. displacement, or for a truck of from 1½- to 3½-ton capacity. Here, power may be taken off the end of the countershaft which projects beyond the end plate of the case and is drilled and tapped for a 7/8-in. S.A.E. thread. In this model the shift bars extend the entire length of the case with bearings at each end, while in that shown in Fig. 169, the bars extend only to the shift forks. Instead of spring plungers, a steel ball with a spring above is employed, for locking the bars in their respective positions. Figure 173 is a control set suitable for the last trans-

mission. It is attached to an extension of the clutch end plate as seen from Fig. 102, which shows the clutch made to be furnished with this control set and the transmission last described.

Figure 174 is a cross sectional view and an end elevation of the transmission manufactured by the Durston Gear Co., suitable for passenger cars weighing up to 3,000 lb., with 30-hp. motor, or for trucks up to 1-ton capacity. The clutch gear shaft and the transmission shaft are running on ball bearings, held against end thrusts in both directions by clamping their upper as well as their lower races. The countershaft is rigidily held in the case; the countershaft gears are all made in one unit, a bronze bushing being pressed into each end and revolve on the shaft.

The bearing retaining cap *b* is made of cast iron and has a spiral oil groove shown at *d*, to lead the oil, which tends to run out, back to the case. The bearing retainer *c* is a malleable casting; *t* is a fabric ring forming a friction surface for the clutch brake.

The countershaft unit gears, are made of 3½ per cent nickel steel, (S.A.E. 2320). The countershaft is made of 1020 carbon steel, hardened and ground to .9975 to .9998 in. Lubrication is provided through the oil holes *O*, noted on the gear unit, and the oil groove of the bushing Fig. 174*a*. Attention is drawn to the dimension of the bushing bore; at first it is made, .960 to .965 in. and is enlarged and reamed to the correct size after it is pressed into the gear. The note on the top provides however that bushings to be supplied as spare parts must be immediately finished to the correct size. The reason is obvious—the garage or service station making repairs, has not, as a rule, the facilities for finishing the bushing. These are mere details, but the engineer must think not only of the first production, but follow in his mind the product through to the consumer, until the parts wear, and then provide for efficient renewals.

Fiber washers ground to .128 to .122 in., are placed at the ends of the countershaft to take the slight end thrust which may arise, and at the same time be noiseless; they are marked *e-e* in the assembly.

A bronze bushing is also provided at *f* for the pilot of the primary shaft. The shaft has four splines, a section being shown in Fig. 174*b*. Oil grooves are placed on the pilot where it revolves in the bushing. The shaft is made of chrome-nickel steel, S.A.E. 3140, hardened and ground. Figure 174*c* is a detail of the high

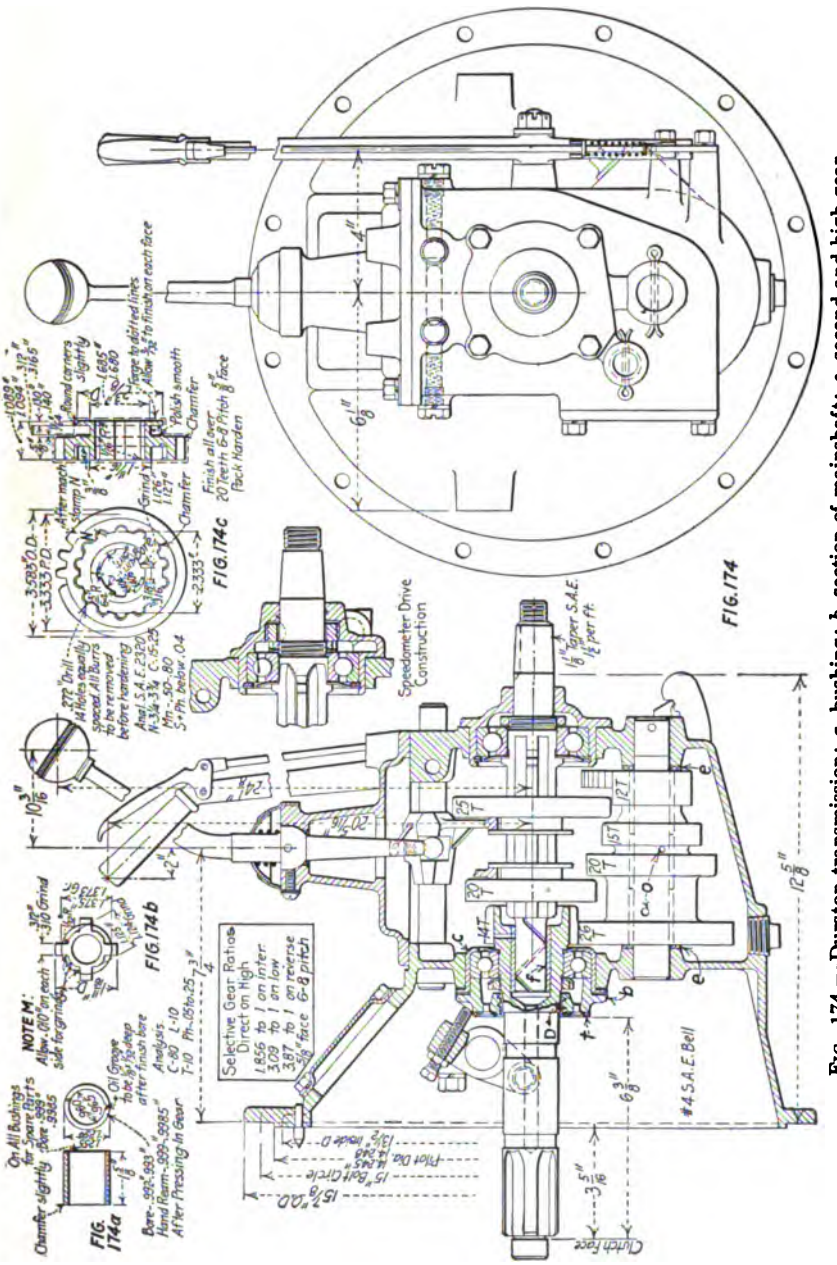


FIG. 174
FIG. 174a
FIG. 174b
FIG. 174c
Speedometer Drive Construction
Selective Gear Ratios
Direct on High
1.856 to 1 on inter
3.09 to 1 on low
3.87 to 1 on reverse
6-8 pitch
#4 S.A.E. Bell
Clutch face
NOTE M:
Allow .010" on each side for grinding
On All Bushings for Spare Parts
Chamfer slightly
Bore-.993/.991"
Hand Ream-.999/.9985"
After Pressing In Gear
Analysis
C-.60 L-.10
T-1.0 Ph-.05b-.25
Finish all over
20 Teeth 6-8 Pitch
Face Harden
Polish smooth
Chamfer
Grind
Chamfer
Flange to dotted lines
Allow 1/32 to finish on each face
Spread All Burrs
Before Hardening
Drill
Holes Equally
Spaced
Clamp
After machining
Radial corners slightly
Oil Groove
To be 1/16 deep
after finish bore
Grind
Grind
Grind

and intermediate sliding gear, giving all the dimensions and remarks required for its manufacture. The groove for the shift fork prong is polished smooth and made about .012 larger than the prongs of the fork.

Figure 175 is an amidship type of transmission manufactured by the Covert Gear Co. suitable for a car having a motor with a maximum torque of 3,000 lb.-inches. It is mentioned in Chap. V, of Vol. I, Engines, that a torque of about 6¾ lb.-inches is obtained per cubic inch of piston displacement. Hence if it is desired to figure out the size of the cylinders, or the total volume

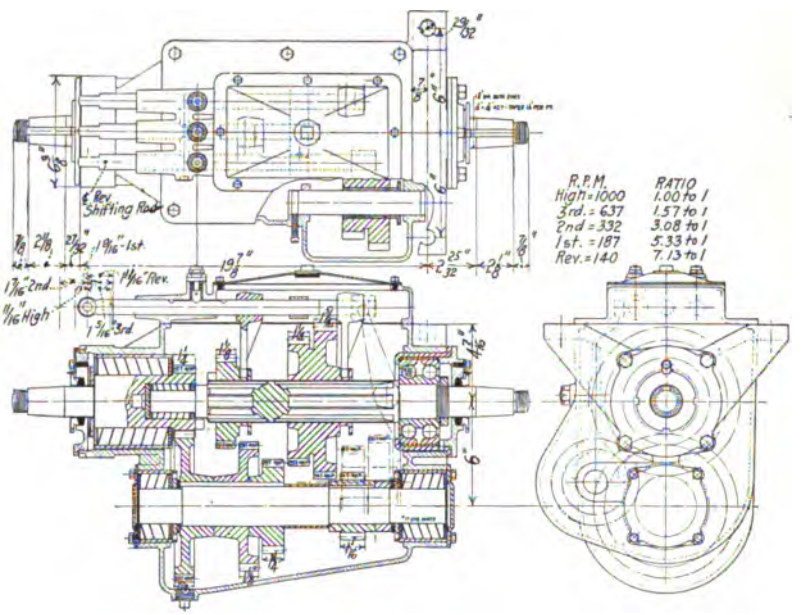


FIG. 175.—Transmission of Covert Gear Co., amidship type.

swept by all the cylinders, it is only required to divide 3,000 by 6¾, which equals 450 cu. in. By dividing this sum by the number of cylinders, the volume swept by each cylinder is found. In a four-cylinder motor the piston displacement per cylinder would then be 112.5 cu. in. which would approximately correspond to a bore of 4¾ in. and a stroke of 6 in. In the smaller models made by this company, the mainshaft has four splines, in the larger models six. In these transmissions the high gear is established by a jaw clutch, the jaws protruding from the clutch gear on one part, and from the sliding gear on the other

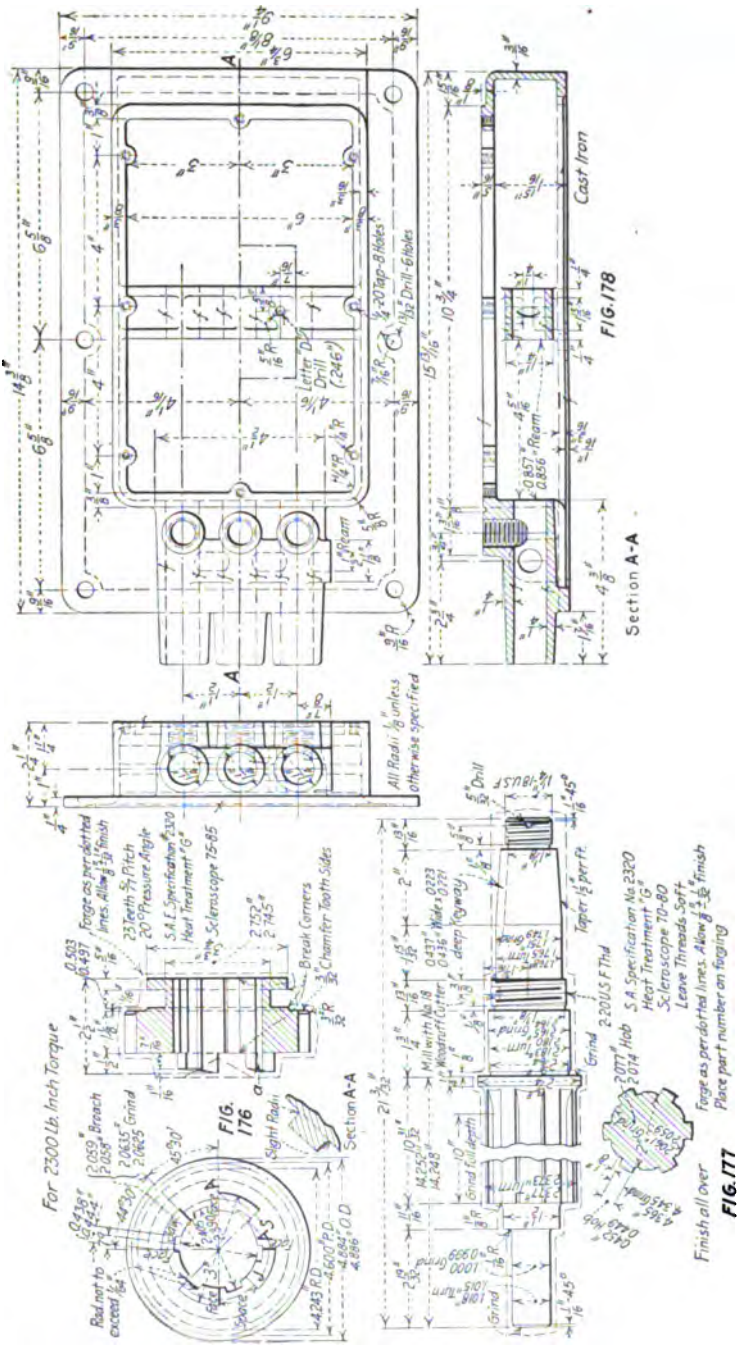


FIG. 176.—Third and high gear. FIG. 177.—Mainshaft. FIG. 178.—Shifter box. (*Covert Gear Co. Transmission.*)

part. Figure 176 shows the high and third sliding gear of a transmission suitable for a motor torque of 2,300 lb.-inches. The jaws or prongs *a* are plainly seen. The gears are made of 2320 S.A.E. steel carbonized (heat treatment *G*), see Vol. I, Engines, for S.A.E. steels and their heat treatments. Figure 177 is the transmission shaft for the sliding gears; it is made of the same material as the gears and heat treated in the same manner, with the exception of the threads which are left soft.

The thread on the end is of course used for attaching it with the universal joint of the propeller shaft, the other thread, for clamping the lower ball race of the bearing.

The face width of the gears for a motor torque of 1,350 lb.-inches is $1\frac{3}{16}$ in. For other sizes manufactured by the Covert Gear Co. the faces of the gears are as follows:

Torque, pound-inches	Face width low gears, inches	2nd, inches	3rd, inches	4th, inches	Reverse, inches
1,850	1	$\frac{15}{16}$	$\frac{15}{16}$	.....	1
1,900	$1\frac{3}{16}$	1	$\frac{15}{16}$	$\frac{15}{16}$	$1\frac{1}{16}$
2,300	$\frac{15}{16}$	1 ¼	1 ⅛	1 ¼	
3,000	$\frac{15}{16}$	1 ¼	1 ⅛	1 ¼	

In giving the width of gears, the sliding gears of the transmission shaft were considered, except for the high gears, (third or fourth as the case may be) where the width of the clutch gear was given. The countershaft driving gear is made slightly narrower, (about ⅛ in.), in the transmissions manufactured by this company. When three speeds are provided, the reverse is usually in constant mesh with the smallest gear of the countershaft, and the reverse is obtained by moving the low speed sliding gear to the right. With four-speed transmissions, the reverse consists frequently of two gears, or of one wide gear (depending as to whether a higher ratio is desired on the reverse than on the low) and the reverse speed is obtained by sliding the reverse gear into mesh with the low speed gear of the main shaft, (see Fig. 175).

Figure 179 is the control set for the last transmission. This is shown fastened upon or carried by a cross shaft or tube of the frame; in their latest model this company mounts the control set on the clutch housing. The principal dimensions are given

in the drawing. Figure 178 is a detail drawing of the shifter box, made of cast iron; for passenger cars this is often made of aluminum, but for trucks cast iron serves the purpose, as a small increase in weight is unimportant, and it is less expensive than aluminum. The wall thickness is $\frac{3}{16}$ in. Even with aluminum it is rarely possible to make the wall thinner than $\frac{5}{32}$ in. on account of the difficulty in pouring the metal when the wall is made thinner. Though the drawings may specify a lesser

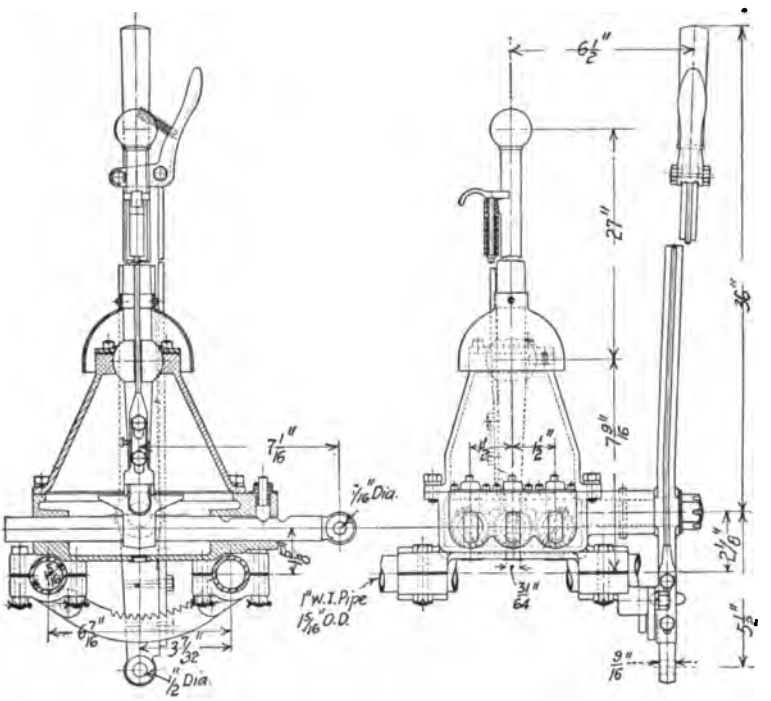


FIG. 179.—Control set. (*Covert Gear Co.*)

thickness and the patterns made accordingly, the finished castings received from the foundry will nevertheless, ordinarily, be nearer $\frac{5}{32}$ or $\frac{3}{16}$ in. Of course, small, simple castings may be made thinner, but where a great deal of coring is necessary this is not easily obtained with the usual foundry processes. The shift box illustrated is used for a unit power plant type of transmission, suitable for a motor torque of 2,300 lb.-inches.

Figure 180 is the shifter box or transmission case cover of the Reo passenger car, having a wall thickness of $\frac{5}{32}$ in. This is

made of cast iron; all the dimensions are given. This car has a six-cylinder motor with a bore and stroke of $3\frac{3}{16}$ by 5 in., that is to say, a piston displacement of 239 cu. in.; the car weight is about 3,000 lb.

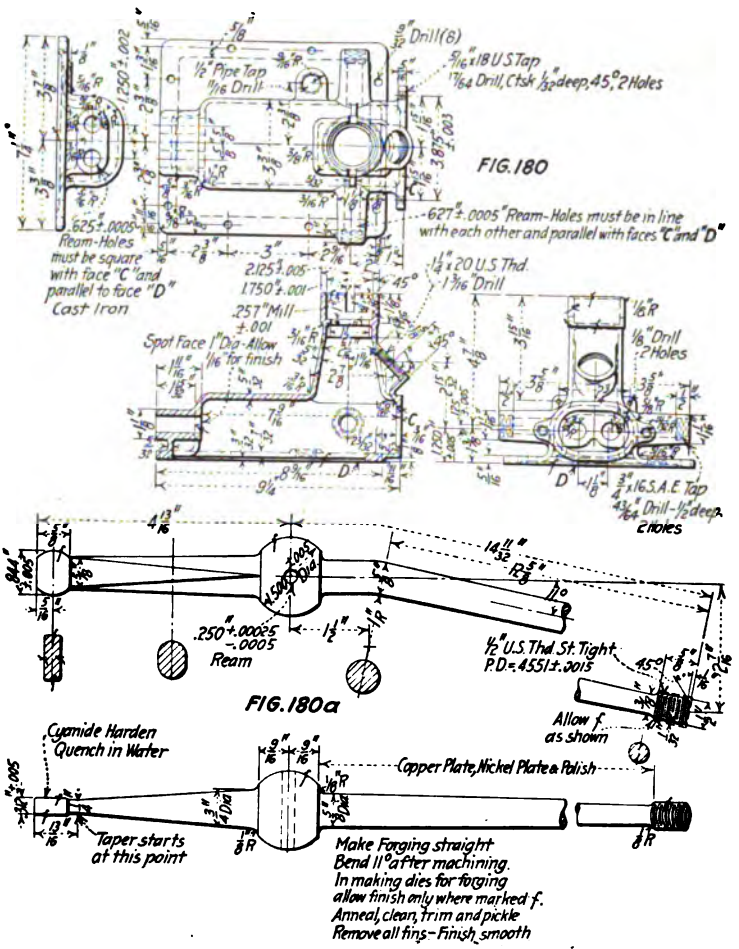
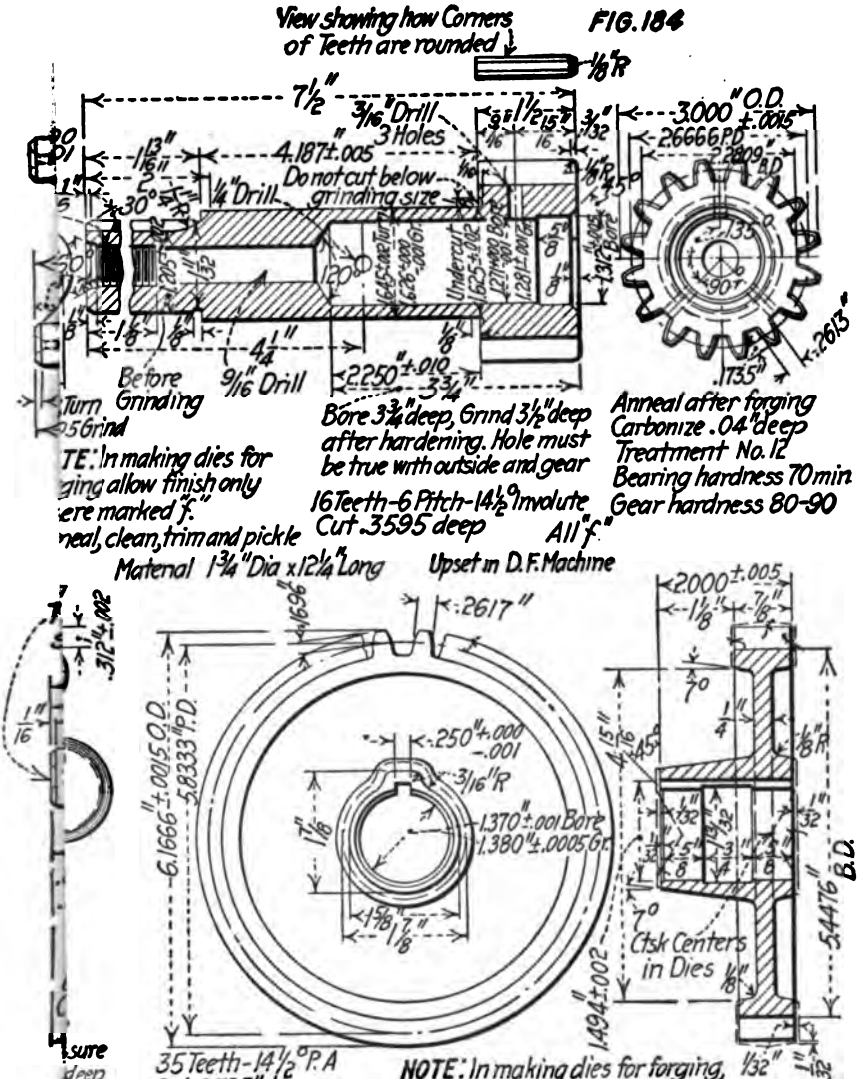


FIG. 180
FIG. 180a
Drill(8)
½" Pipe Tap
11/16 Drill
Ream-Holes must be square with face "C" and parallel to face "D"
Cast Iron
Spot Face 1" Dia-Allow 1/16 for finish
.627"±.0005" Ream-Holes must be in line with each other and parallel with faces "C" and "D"
1/16" Drill 2 Holes
.250"+.0005 -.0005 Ream
½" U.S. Thd. St. Tight
Allow f. as shown
Cyanide Harden Quench in Water
Copper Plate, Nickel Plate & Polish
Taper starts at this point
Make Forging straight
Bend 11° after machining.
In making dies for forging allow finish only where marked f.
Anneal, clean, trim and pickle
Remove all fins-Finish smooth

FIG. 180.—Shifter box of Reo transmission. FIG. 180a.—Gearshift lever.

Figure 181 gives assembly views of this transmission. It is provided with Hyatt roller bearings throughout except for the pilot of the main shaft, which has a plain bearing. The bearings are non-adjustable except in an endwise direction to allow for lining up the gears. The upper rear bearing sleeve *F* is adjus-



38.—Countershaft drive gear (transmission parts of Reo six-cylinder car).

(Facing Page 172.)

table endwise to take up any lost motion. A series of six holes equally spaced around its circumference and $\frac{1}{32}$ in. apart endwise, provide for this adjustment. To make adjustments the shoulder screw *G* is removed and the bearing sleeve driven in or forward until the endwise motion is taken out of the mainshaft. Then the sleeve is turned around until a hole is found to register with that of the shoulder screw, when the latter is screwed in thereby locking the sleeve in position. The countershaft bearings are provided with threaded caps for endwise adjustment. This transmission is placed amidship and supported at four points on the subframe which also carries the motor.

The electric starter is connected to the sprocket *A* by means of a chain, *V* being the starter ratcket and *W* the starter pawl; the speedometer is connected to gear wheel *B*. Oil grooves *O* for the felt washers are cut spirally like a screw, in order to wipe the oil towards the case when the car is running. The shifting mechanism and shift bar locks are seen from the figure on the left; the end *E* of the gearshift lever forces the loose plungers *P* to one side which releases the spring plunger *S* from the bar to be shifted while at the same time permitting the other spring plunger to enter the hole of the other bar more deeply, to prevent its shifting out of place until the end *E* of the gear shift lever is returned to its neutral position.

Figure 180*a* is a detail of the gearshift lever. The manner of retaining this lever in the gear case cover can be seen from the assembly drawing. Figure 182 shows a detail of the shift bar, (low and reverse) and Fig. 183 a detail of the shift fork. The shift forks are provided with a screw thread whereby they are held to the shift bars as shown. By these means the position of the shift forks may be changed endwise to adjust the position of the various sets of gears.

Figure 184 is a detail of the clutch gear and its shaft, Fig. 185 the main shaft, and Fig. 186, the countershaft. Note how the latter is stepped, to permit the center gears to slide easily over the end of the shaft for assembling. The sliding gears are provided with four spline keyways while the countershaft gears are attached by means of Woodruff keys. The gears are $\frac{7}{8}$ in. in width, except the clutch gear, which drives the countershaft as well as forms a clutch for the direct drive, which is $1\frac{1}{2}$ in. wide, and the high and intermediate sliding gear, (Fig. 187), which is $1\frac{3}{8}$ in. wide. The dimensions and tolerances are marked and

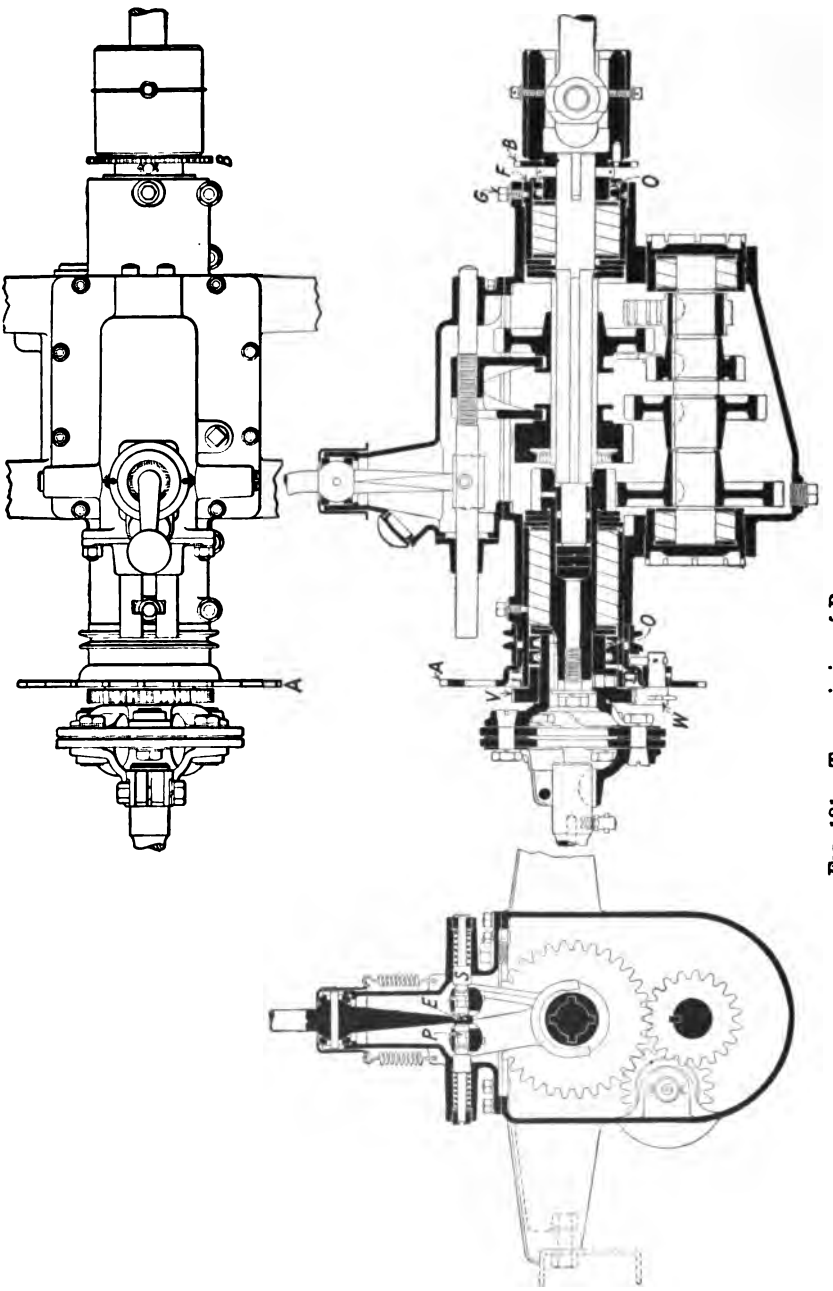


FIG. 181.—Transmission of Reo passenger car.

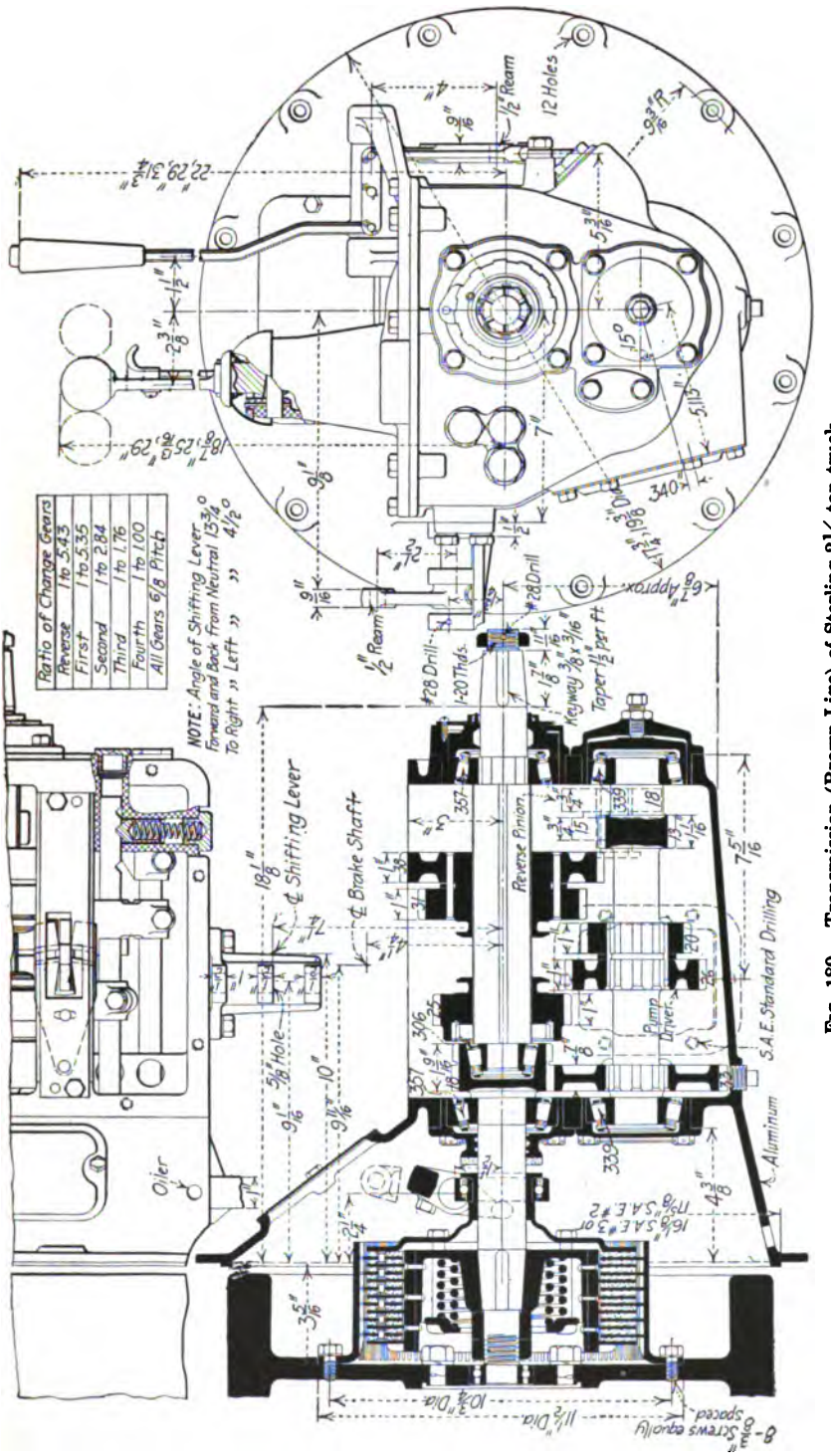


FIG. 189.—Transmission (Brown-Lipe) of Sterling 2½-ton truck.

are self explanatory. Figure 188 is the countershaft drive gear; all the gears are carbonized as indicated and have a minimum scleroscope hardness as noted; they have standard involute teeth with a pressure angle of 14½°.

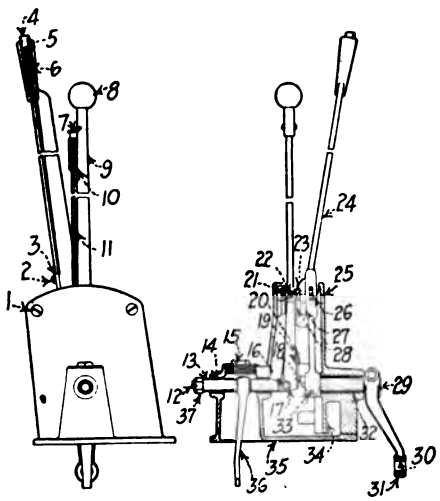


FIG. 190.—Sterling truck gearshift control.

Figure 189, is the transmission (Brown-Lipe) used on the Sterling 2½-ton truck, which is made in either three or four speeds forward. The transmission shaft has a square fit and all the bearings are of the Timken roller type. The gear shift control parts are either as shown in the last drawing or like that of Fig. 190; the numbers refer to the names of the various parts.

Name of Part	Name of Part
1. Gear shift quadrant bolt.	*21.* Gear shift poppet.
2. Gear shift brake pawl screw.	*22.* Gear shift poppet spring.
3. Gear shift brake pawl.	*23.* Brake plain quadrant.
4. Gear shift brake pawl rod button.	*24.* Brake lever.
5. Brake pawl rod button pin.	*25.* Gear shift cover plate.
6. Brake pawl spring.	*26.* Brake quadrant.
7. Gear shift reverse gear rod push button.	*27.* Brake and plain quadrant spacer.
8. Gear shifting lever handle.	*28.* Brake plain quadrant spacer.
9. Gear shifting lever.	*29.* Brake hand lever shaft:
10. Gear shift reverse gear rod push button spring.	*30.* Brake hand lever bushing.
11. Gear shift reverse gear rod.	*31.* Brake hand offset lever.
12. Gear shift hand lever shaft.	*32.* Brake hand lever shaft bushing.
13. Gear shift hand lever shaft sleeve.	*33.* Brake hand lever screw.
14. Gear shift hand lever shaft bushing.	*34.* Gear shift cover plate screw.
15. Gear shift cover cap.	*35.* Gear shift cover.
16. Gear shift lever bolt nut.	*36.* Gear shifting finger.
17. Gear shift lever bolt.	*37.* Gear shift hand lever shaft nut.
18. Gear shift reverse gear stud pin.	*38.* Brake hand lever shaft grease cup.
19. Gear shift reverse gear lock.	*39.* Brake hand offset lever key.
20. Gear shift reverse gear lock stud.	*40.* Brake hand lever key.
	41. Gear shift lever set, complete.

The first and second speed sliding gears are carried on a gear carrier, (Fig. 191), to which the said gears are riveted as shown in

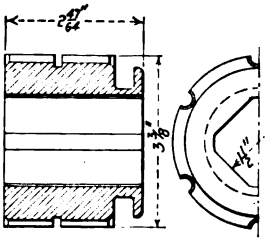


FIG. 191.—Gear carrier.

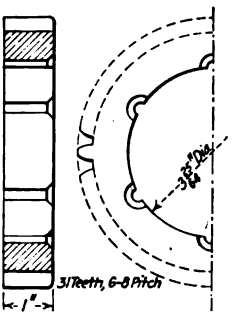


FIG. 192.—Second speed gear.

the assembly. Figure 191 is an illustration of one of these gears. The countershaft gears are riveted to the shaft like those shown in (Fig. 163). The width of the gears is 1 in., except the counter-

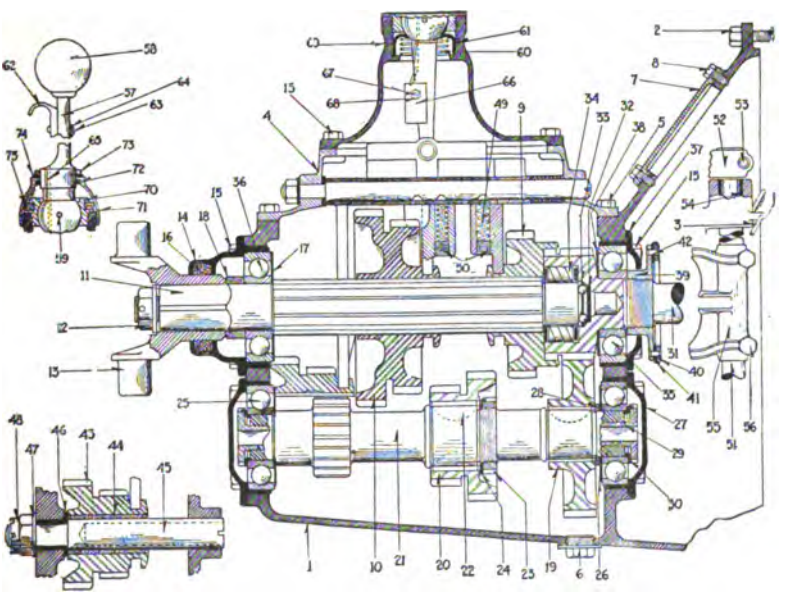


FIG. 193.—Transmission of White 5-ton truck.

shaft drive gear which is only $\frac{7}{8}$ in., and the reverse gears, shown dotted, which are $\frac{3}{4}$ in.

Figure 193 is the transmission, forming a part of the unit power plant of the White 5-ton truck. The shafts run on ball

bearings with the exception of the main shaft pilot which is provided with Hyatt rollers; at the pilot end a ball thrust bearing is located.

Ref. No.	Number Per Car	Name
1	1	Transmission case (lower half).
2	10	Transmission case bolts.
	2	Transmission case bolts.
3	1	Transmission case oil tube.
4	1	Transmission case (upper half).
5	10	Transmission case cover screws.
	10	Transmission case cover screw lock washers.
6	1	Transmission case drain plug.
	1	Transmission case filler plug.
7	1	Transmission hand hole cover.
8	6	Transmission hand hole cover screws.
	6	Transmission hand hole cover screw lock washers.
9	1	Transmission sliding gear.
10	1	Transmission sliding gear.
11	1	Transmission sliding gear shaft.
12	1	Transmission sliding gear shaft nut.
13	1	Transmission sliding gear trunnion.
14	1	Transmission sliding gear bearing cage.
15	12	Transmission sliding gear bearing cage cap screws.
	12	Transmission sliding gear bearing cage cap screw lock washers.
16	1	Transmission sliding gear bearing cap felt.
17	1	Transmission sliding gear bearing cap washer (large).
	1	Transmission sliding gear bearing cap washer (small).
18	1	Transmission sliding gear bearing spacer.
19	1	Transmission back gear.
20	1	Transmission back gear.
21	1	Transmission back gear shaft.
22	4	Transmission back gear shaft keys.
23	1	Transmission back gear shaft lock nut.
24	1	Transmission back gear shaft lock nut washer.
25	2	Transmission back gear shaft bearings.
26	2	Transmission back gear shaft bearing washers.
27	2	Transmission back gear shaft bearing cage.
28	1	Transmission back gear shaft bearing spacer.
29	2	Transmission back gear shaft bearing spacer nuts.
30	2	Transmission back gear shaft bearing spacer nut lock washers.
31	1	Transmission clutch gear.
32	1	Transmission clutch gear race.
33	1	Transmission clutch gear race retainer.
	6	Transmission clutch gear race retainer steel balls.
34	1	Transmission clutch gear Hyatt roller bearing (high duty).
35	1	Transmission clutch gear bearing.
36	1	Transmission clutch gear bearing.
37	1	Transmission clutch gear bearing cage.
38	1	Transmission clutch gear bearing cage washer.
39	1	Transmission clutch gear brake flange.
40	1	Transmission clutch gear brake flange facing.
	1	Front half change gear level bracket.
	1	Rear half change gear lever bracket.
	1	Hand hole cover gasket (cork).
40	2	Bearing cage gasket (M.P.).
	1	Transmission case cover gasket (M.P.).
	1	Brake pedal adjusting plug.
	1	Transmission case bottom plug.

Ref. No.	Number Per Car	Name
41	1	Transmission clutch gear brake flange facing cup.
42	6	Transmission clutch gear brake flange rivets (hollow brass).
43	1	Transmission reverse gear.
44	1	Transmission reverse gear bushing.
45	1	Transmission reverse gear shaft.
46	1	Transmission reverse gear shaft bushing.
47	1	Transmission reverse gear shaft washer.
48	1	Transmission reverse gear shaft nut.
49	3	Transmission gear shifter springs.
50	3	Transmission gear shifter spring plugs.
	3	Transmission gear shifter spring steel balls.
51	1	Transmission clutch throw out shaft.
52	1	Transmission clutch throw out shaft lever.
53	1	Transmission clutch throw out shaft lever binder bolt.
	1	Transmission clutch throw out shaft key (No. A Woodruff).
54	2	Transmission clutch throw out shaft bushings.
55	1	Transmission clutch throw out shaft yoke
56	2	Transmission clutch throw out shaft yoke screws.
	2	Transmission clutch throw out shaft yoke screw nuts.
	2	Transmission clutch throw out shaft yoke screw nut lock washers.
57	1	Transmission change gear lever.
58	1	Transmission change gear lever ball.
59	1	Transmission change gear lever pin.
60	1	Transmission change gear lever spring.
61	1	Transmission change gear lever spring washer.
62	1	Transmission change gear lever latch.
63	1	Transmission change gear lever latch screw.
64	1	Transmission change gear lever latch screw washer.
65	1	Transmission change gear lever latch rod.
66	1	Transmission change gear lever block.
	1	Transmission change gear lever block spring.
67	1	Transmission change gear lever block screw.
68	1	Transmission change gear lever block screw nut.
69	1	Transmission change gear lever bracket.
70	1	Transmission change gear lever bracket nut.
	6	Transmission change gear lever bracket screws.
	6	Transmission change gear lever bracket screw lock washers.
71	1	Transmission change gear lever bracket cover.
72	1	Transmission change gear lever bracket cover nut.
73	1	Transmission change gear lever bracket cover nut.
74	1	Transmission change gear lever bracket cover nut washer.
75	1	Transmission change gear lever bracket cover felt.
	1	Transmission foot lever bracket.
	4	Transmission foot lever bracket screws.
	1	Transmission foot lever bracket washer.
	1	Transmission foot lever shaft.
	1	Transmission foot lever shaft bushing.
	1	Transmission foot lever shaft washer.

Note the provisions for preventing oil leaks, washers *17* and *38*, placed on the inner side of the bearings, are clamped by the races next to the shaft, and extend to the upper races which they almost touch. These washers also prevent grit from finding its way to the bearings. The outside bearing cages *14* and *37* form oil reservoirs where oil may collect, should it get through the ball bearings, whence it can drain back again through such ball

bearings, and into the case. The low gear on the countershaft is forged integral; the second and third gears *20* are made in one piece, keyed to the shaft, and are held against displacement by a nut *23* which is provided with a lock nut washer *24*. The constant mesh countershaft gear *19* is held between a shoulder on the shaft and the bearing spacer *28*. Provisions must always be made to prevent gears from shifting, even when they are keyed to the shaft; this is accomplished in various ways as may be

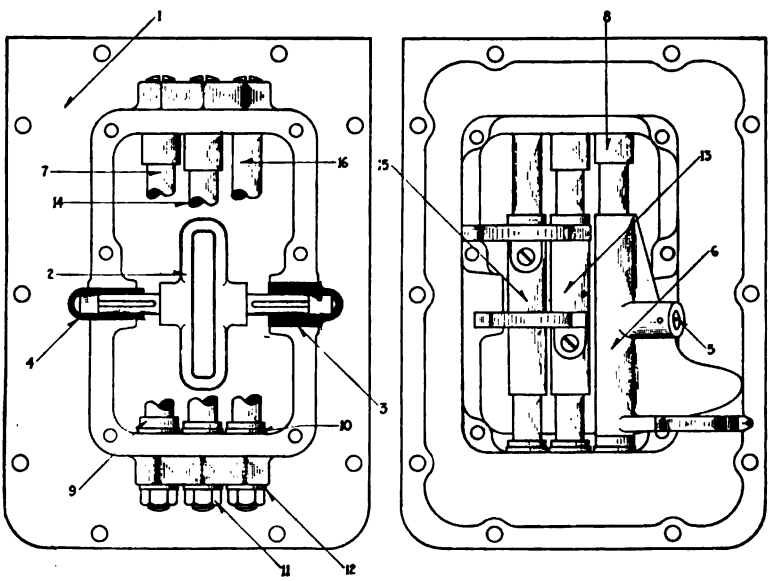


FIG. 194.—Transmission cover of White 5-ton truck.

perceived from an examination of the different designs of transmissions shown in this chapter.

In some designs there is only one bearing in the transmission which supports the clutch gear; when such is the case, the other end of the shank or shaft is supported by a bearing in the clutch or by a pilot bearing in the end of the crankshaft. Figure 194 is the transmission cover assembly; its operation can be understood by referring to the names of the various parts which indicate their functions.

The gear shift lock *2* moves to the right or to the left only, to permit engagement with the various gear shifters *6*, *13* and *15*. Moving these gear shifters forward or backward, by the hand lever, meshes the various sets of gears.

Ref. No.	Number per car	Name
1	1	Transmission case cover.
2	1	Transmission gear shifter lock.
3	1	Transmission gear shifter lock bushing.
4	1	Transmission gear shifter lock bushing.
5	3	Transmission gear shifter lock plugs.
	3	Transmission gear shifter lock springs.
	3	Transmission gear shifter lock steel balls.
6	1	Transmission reverse gear shifter.
7	1	Transmission reverse gear shifter latch bar.
8	1	Transmission reverse gear shifter stop.
9	1	Transmission reverse gear shifter stop.
10	2	Transmission reverse gear shifter stop washers.
11	3	Transmission shifter latch bar nuts.
12	3	Transmission shifter latch bar nut washers.
13	1	Transmission direct and high speed gear shifter.
14	1	Transmission direct and high speed gear shifter latch bar.
15	1	Transmission first and second speed gear shifter.
16	1	Transmission first and second speed gear shifter bar.
	1	Transmission reverse gear shifter stop.

The Pierce-Arrow 2-ton truck transmission is illustrated in Fig. 195; it is of the amidship type. The gear ratios are: On the low, 5.2 to 1; on the second, 2.7 to 1; on the third, 1.6 to 1; on the fourth 1 to 1; and on the reverse 7.3 to 1, that is to say when the motor shaft makes 1,000 revolutions per minute, the primary or main shaft connected to the propeller shaft would make 1,000 revolutions per minute on the direct or fourth speed, 625 $\left(=\frac{1,000}{1.6}\right)$ on the third, 371 on the second, 192 on the first, and 137 revolutions on the reverse. Section *A-A* shows the reverse idlers and the manner in which they are shifted. All the other parts and their functions are readily understood from the description of the previous designs. In the drawing at the left top, *a* shows a stop for the reverse pinned to the shift bar; *b* shows plungers, which are used in addition to the regular spring plungers, to prevent the shifting or dislocation of more than one shift bar at one time. This is the latest product of this well known company and its careful study is recommended to students.

Figure 196 is the transmission of the Pierce-Arrow six-cylinder passenger car. The second and third speed gears of the countershaft are equipped with spring pawls to facilitate the shifting

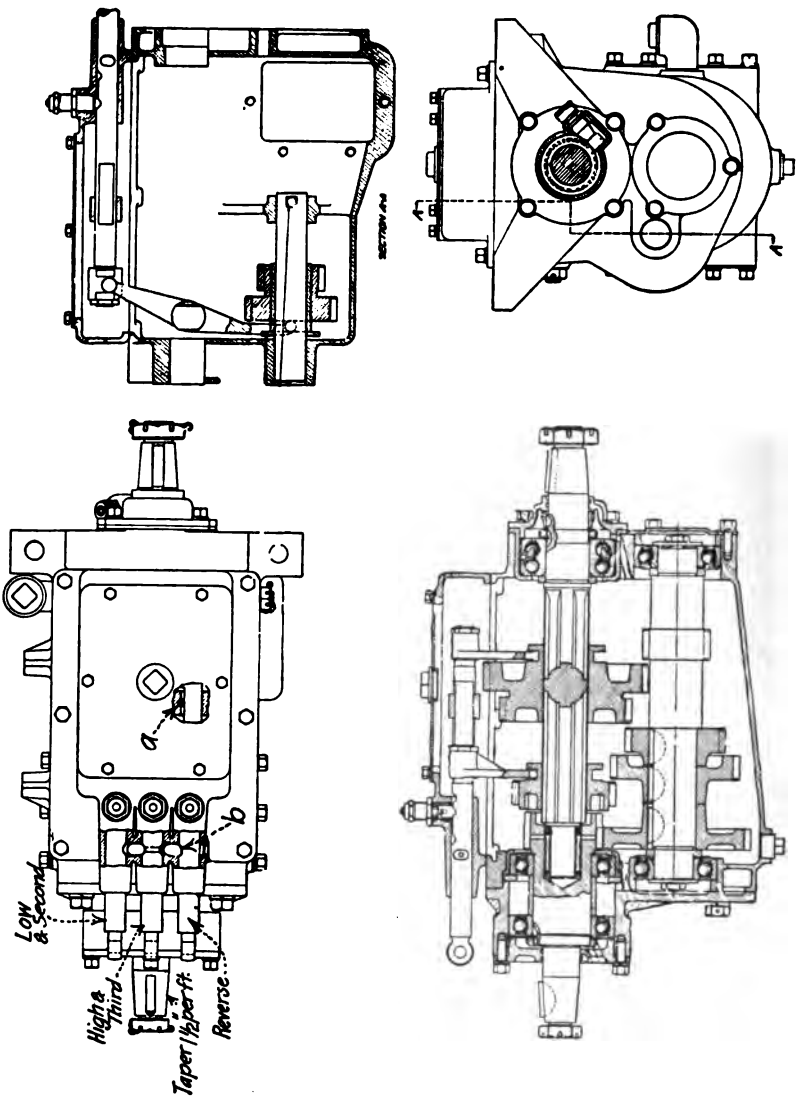


FIG. 195.—Transmission of Pierce-Arrow 2-ton truck.

of gears. These spring pawls cause there gears to act as a free wheel in one direction, and this permits shifting with extreme ease from the high to the second or the third; the only function sacrificed is that on the second or third speeds the engine cannot be used as a brake. Note the spiral in the clutch gear bearing to prevent oil leaks.

Figure 197 is a section of the countershaft with all the gears in place. The countershaft is a drop forging. The sectional view *A-A* shows the arrangement of the pawls.

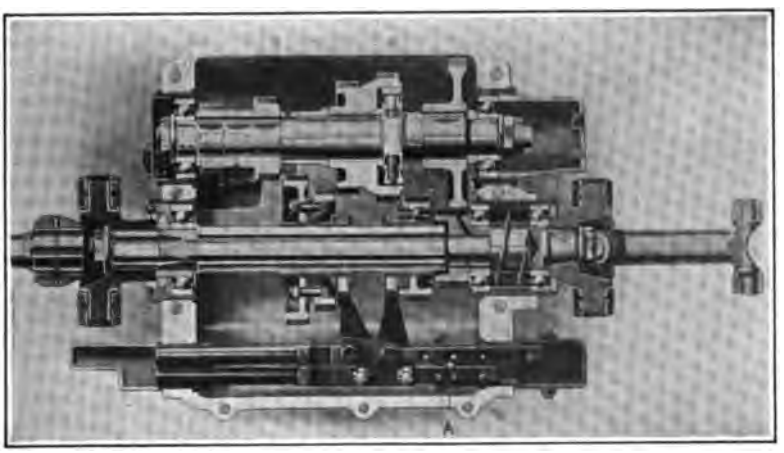
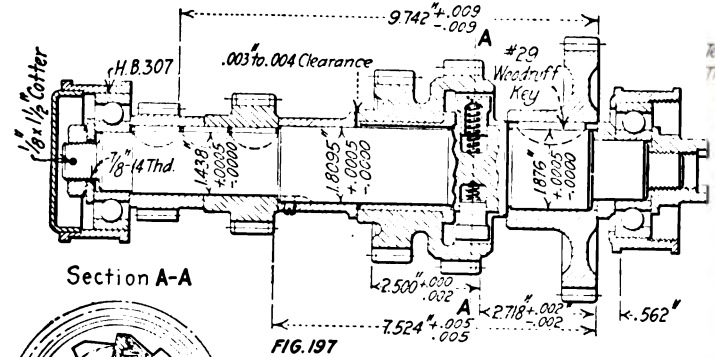


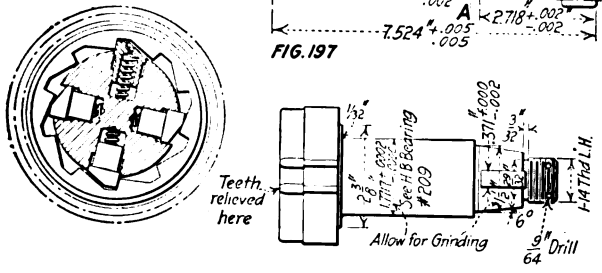
FIG. 196.—Transmission of Pierce-Arrow six-cylinder passenger car.

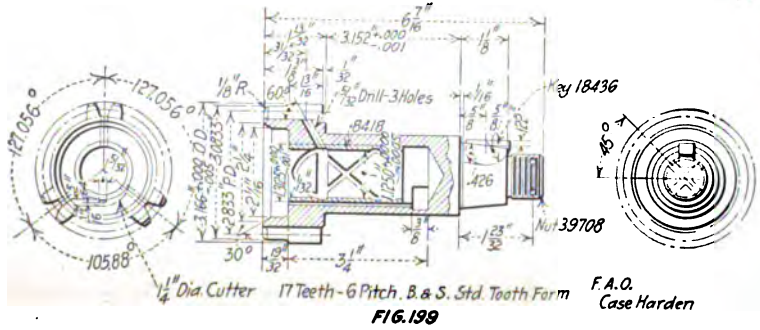
Figure 198 is a detail of the second and third speed gears of the countershaft which engage the spring pawls. As may be noted there are 10 internal notched teeth on these gears, and this enables two pawls to come into engagement simultaneously, (see Section *A-A* Fig. 197).

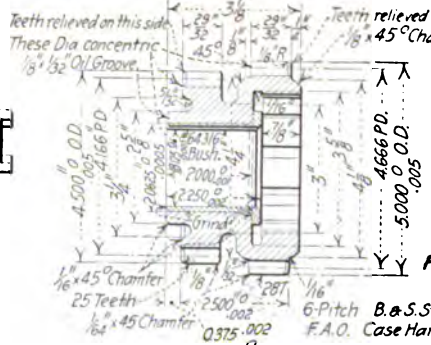
Figure 199 is a detail of the clutch gear with its shank. The gears of this transmission are six pitch and have the Brown & Sharp standard tooth from, *i.e.*, they have a pressure angle of 14½°. The end of the teeth are somewhat relieved to mesh with the internal teeth of the third speed sliding gear, for the direct drive. The pilot bearing is a plain bushing with oil grooves as seen from this illustration.

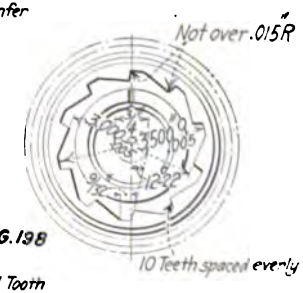
Figure 200 is the transmission shaft made of 1⅞-in. diameter stock. It has six splines for the sliding gears and 10 splines for the universal joint hub.

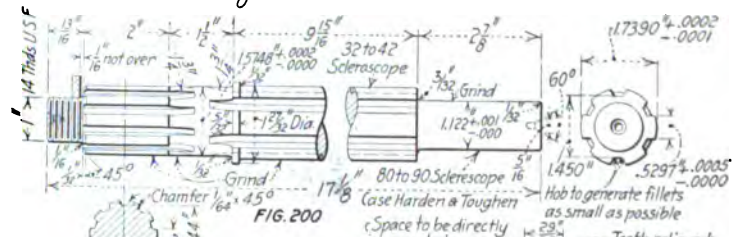












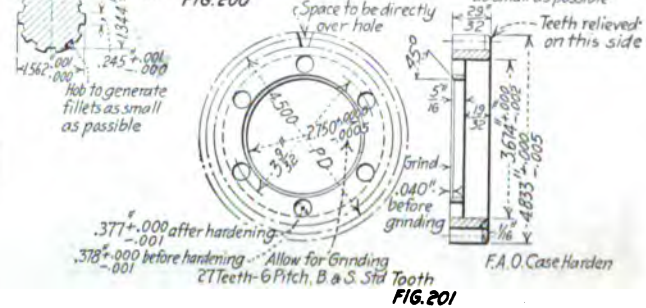

Space to be directly over hole
Teeth relieved on this side
.1562" +.001 -.000
.245" +.001 -.000
Hob to generate fillets as small as possible
4.500
2.750" +.0000 -.0005 P.D.
Grind .040" before grinding
3.664" +.002 -.005
.377" +.000 -.001 after hardening
.318" +.000 -.001 before hardening
Allow for Grinding
27 Teeth-6 Pitch, B. & S. Std Tooth
F.A.O. Case Harden
FIG. 201

Fig. 197.—Countershaft. **Fig. 198.**—Second and third speed gears. **Fig. 199.**—Clutch gear. **Fig. 200.**—Transmission shaft. **Fig. 201.**—Second speed gear (Pierce-Arrow six-cylinder car).

Figure 201 is a detail for the second speed sliding gear which is attached to a sliding boss by means of six bolts as seen from the assembly.

The change speed control is illustrated in Fig. 202. The lever *A* works in a gate having two longitudinal slots and a cross slot, corresponding to the neutral position through which the lever can be moved to engage in either longitudinal slot. The gate is fixed to a tube *B* on the other end of which is the lever *C*, which engages with either of the shift bars *D* or *E*. When the gears are

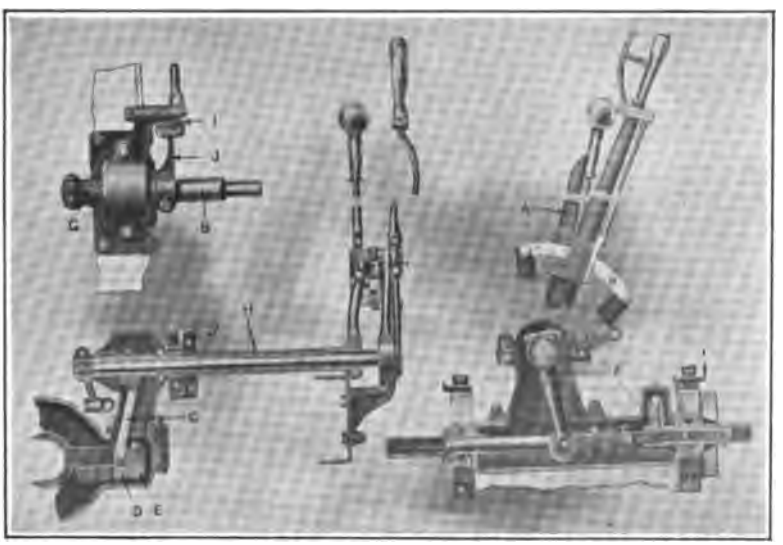


FIG. 202.—Pierce-Arrow change speed control (passenger car).

in full mesh the bars are locked by means of a ½-in. ball *F*, pressed into the notches of the bar by a spring above it. It is impossible to get two gears into mesh at the same time, due to the locking pin *A* (Fig. 196), which permits either of the two bars to move only after the other has been locked. Another locking device prevents moving the shifting lever until the clutch is disengaged, and the clutch cannot be engaged until the gears are in proper mesh. The interlocking lever is locked by a bolt *I* connected to the clutch lever, so that when the clutch is disengaged, this bolt is withdrawn from quadrant *J*.

There has been a tendency lately to provide a greater speed range for truck transmissions in order to permit running the motor at its most efficient speed under all conditions. Examples

of this practice are the latest trucks manufactured by the General Motors Truck Co. In their transmissions are two pairs of constant mesh gears, *i.e.*, the clutch gear shaft *17* has two gears, *6* and *8*, (Fig. 203), in rigid connection with it, the former being made integral with the shaft while the latter is keyed to it by four keys as shown. The countershaft *18* has two free driving gears, *7* and *9*, either of which can be positively connected to it

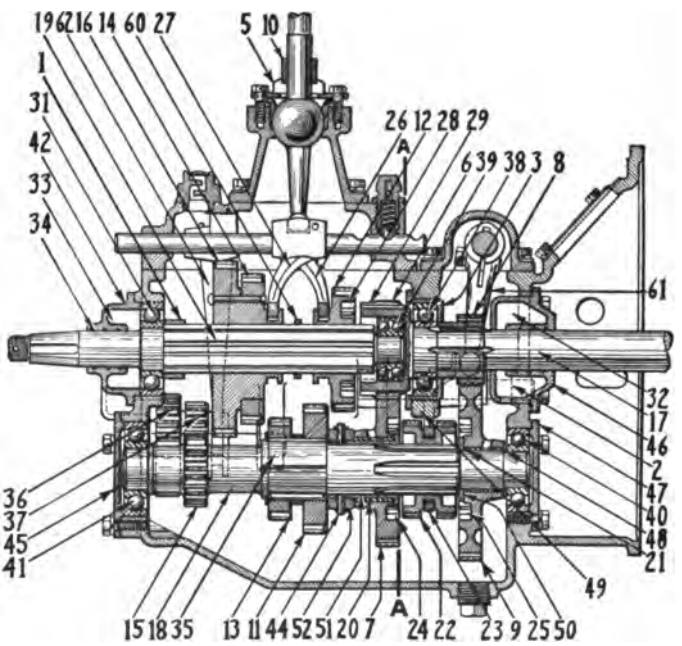


FIG. 203.—Transmission of General Motors Truck Co. (seven forward speeds and two reverse).

by a positive tooth clutch *22*, which is splined to the countershaft and is actuated by shift fork *23*, operated by a separate lever *61*. By throwing this clutch to the right or the left it will engage the larger or smaller countershaft drive gear respectively. (The rest of the transmission functions like the conventional designs.) In this manner, it is possible to obtain two different speeds on the low gear, the second, the third, and on the reverse; these speeds with the fourth or direct, give seven speeds forward and two reverse.

Figure 204 gives a view of the left side of the case, partly

open, disclosing on the left, the shift fork and lever operating the positive clutch previously referred to.

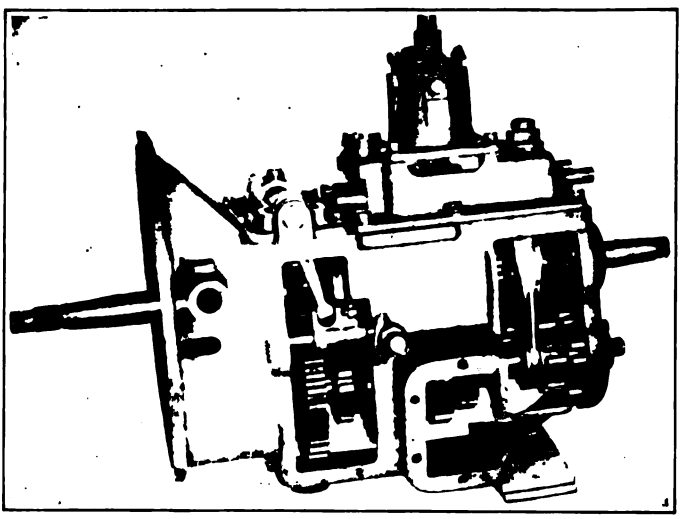


FIG. 204.—General Motor's Truck Co.'s transmission.

In order to obviate the necessity of shifting gears in and out of mesh, which in the beginning of the automobile industry was the cause of much trouble, different manufacturers have at vari-

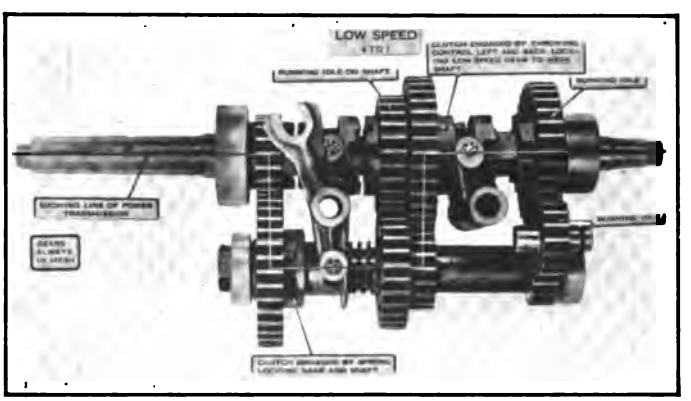


FIG. 205.—Cotta transmission, used on Acme truck.

ous times developed designs which maintained the gears always in mesh, and employed independent clutches to lock the gears to the shafts. Such a design is illustrated in Fig. 205, which is the

Cotta transmission as used on the Acme trucks. All the gears are mounted on roller bearings; at the side of the sliding gears and the countershaft drive gear, and forming a part thereof, are a set of jaw clutches. On the six-spline main shaft, are correspond-

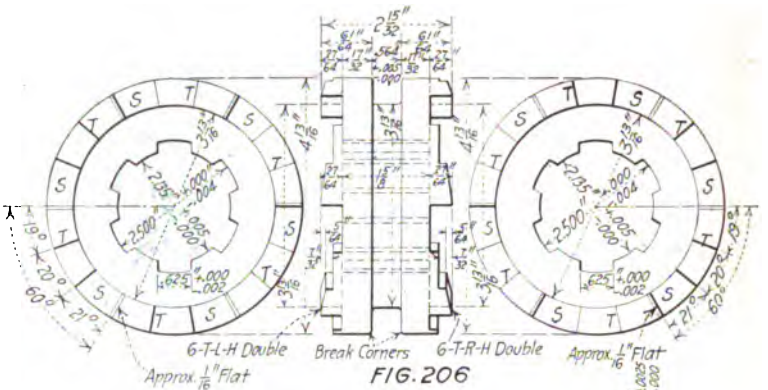


FIG. 206

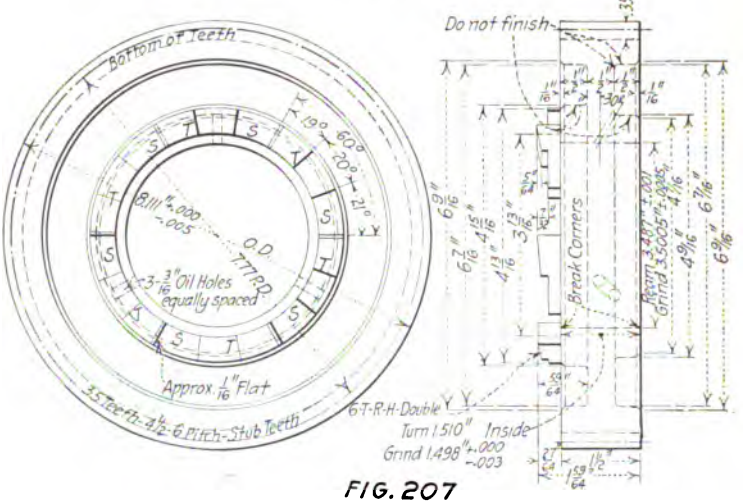


FIG. 207

FIG. 206.—Sliding clutch. FIG. 207.—Second speed gear (Cotta transmission).

ing double sliding clutches, by means of which any of the speed change gears may be locked to the shaft. On high speed (direct drive) the clutch gear shaft and the mainshaft are locked together and the clutch on the countershaft is disengaged; in this position the countershaft with its gears, and all the gears on the transmission shaft, are idle, *i.e.*, they are not running. When the

sliding clutch is returned from the direct drive to the neutral position, the countershaft clutch automatically engages the countershaft driving gear and keeps it there during all the other speeds, by means of the spring seen behind it.

Figure 206 is a detail of the sliding clutches of the main shaft (for first and second gear) suitable for a 7-ton truck, and Fig. 207 a detail drawing of the second speed gear, which show the dimensions and shape of these clutch jaws and the gear width. The gears and clutches are made of chrome-nickel steel, and they ride direct on rollers located inside the gears.

Transmission Cases.—The transmission case, frequently called the gear case, or gear box, is the housing which contains and supports the shafts with their gears by which the various gear changes are effected to obtain different car speeds. Frequently the gearshift hand lever and the brake hand lever are also attached to the case. Gear cases are made of cast iron or aluminum, depending on whether cost or lightness is the first consideration. In the design of gear boxes rigidity is of prime importance. Both wear and noise, may be due to springiness in the case or the shafts. The shorter a beam the greater is its strength and rigidity and the less is its flexure. Gear boxes, should, for these reasons, be made no longer than necessary.

To accommodate the gears, ample clearance, should be allowed, not less than ¼ in.; this refers more especially to the space around the side where the case bulges out to house the gears.

The case is usually cast in one piece with an opening on the top for the cover; when the unit power plant construction is employed, it is also open on the front end, which is enlarged to house the clutch. Sometimes the gear case is bolted to the clutch housing.

A large plug should be provided at the bottom for drainage. If the plug hole is less than 1 in. in diameter it is difficult to drain the case when the grease or oil has become heavy or thick. A filler should be provided at the side of the case, which may also serve the purpose of level gauge, so that the gear case cannot be filled higher than to a given level. In the Class B engine the filling orifice is brought out well to one side, to enable a person to fill the case from the side of the truck. The importance of a vent pipe was mentioned before, its absence being frequently responsible for oil leaks. The bearing end plates should be provided with packing gaskets, for it is here that leakages mostly occur. Vent pipes prevent the formation of pressure in the case

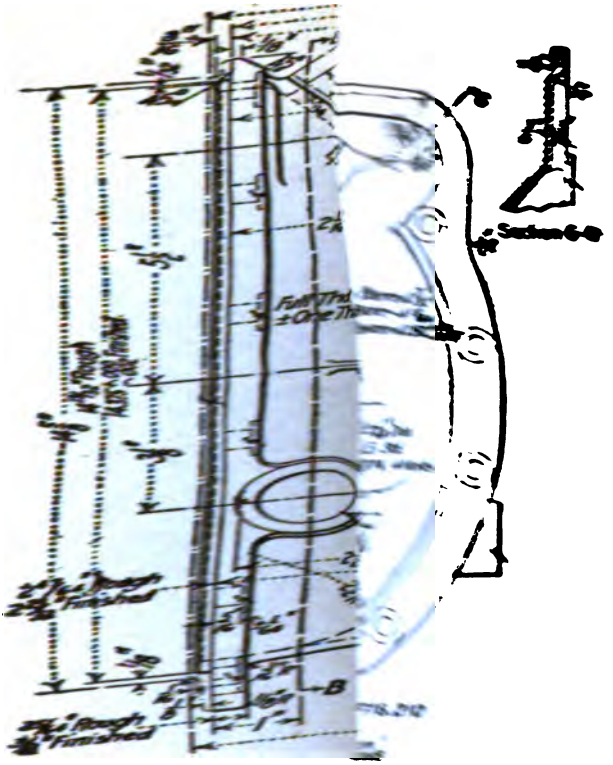
(due to the expansion of air arising from a rise in temperature) and this diminishes the tendency for leaks.

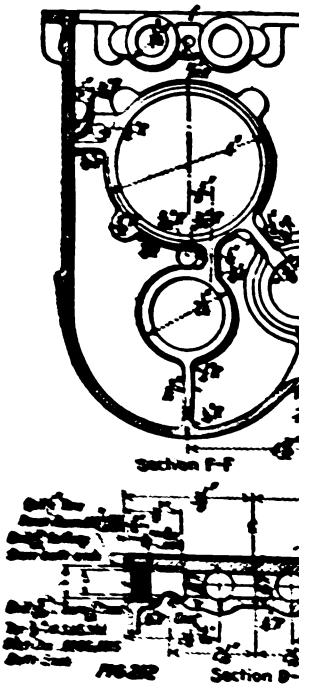
As can be noted from the various designs, the bosses which support the bearings are reinforced by ribs, running vertically as well as horizontally. The bearing retainers, at least one of them, must be made of sufficient size to permit the withdrawal of the shaft from the end; when the countershaft and the clutch gear shaft are provided with an integral gear, they too must pass through the bore in the end of the case. When the brake lever and foot pedals are attached to the case, the bosses holding them, or their shafts, must be well ribbed to insure rigidity.

The transmission case should be designed to present a neat exterior; this often necessitates placing the ribs inside, where they do not interfere with the working mechanism. It is advantageous to have all the bosses on the same plane or level for the sake of appearance and also because it saves "setting-up" operations in the shop, as one machining operation will finish all the surfaces on one side that requires to be finished.

Wherever screws are threaded in aluminum, which often are unscrewed, trouble, is liable to occur, hence the best practice is to employ bolts, or studs permanently fastened into the aluminum with nuts on the outside. The length of tapped holes should not be less than from 2 to 2¼ times their diameter, as aluminum is soft and the threads are liable to tear out. When bolts are used, arrangements must be made to prevent heads or nuts from dropping into the case if they should become loose or when dismantling for repairs.

Transmission Case Suspensions.—In attaching the transmission case to the frame provisions must be made to prevent frame distortions from being transmitted to the gear box and to produce stresses therein. Either the three-point or the four-point suspension is employed in the various designs, whether they be the unit power plant type, where the engine, clutch and gearset are suspended in one unit, or the amidship gearset, which is supported separately; whether supported on a sub-frame, or by the frame side members direct. Sometimes, especially with the amidship type, the gear case is suspended from cross members from above, at other times their supports rest upon cross members or on the frame side members. Great attention should be paid to the gear case supports; they should be properly strengthened where they unite with the rest of the case or the housing proper.





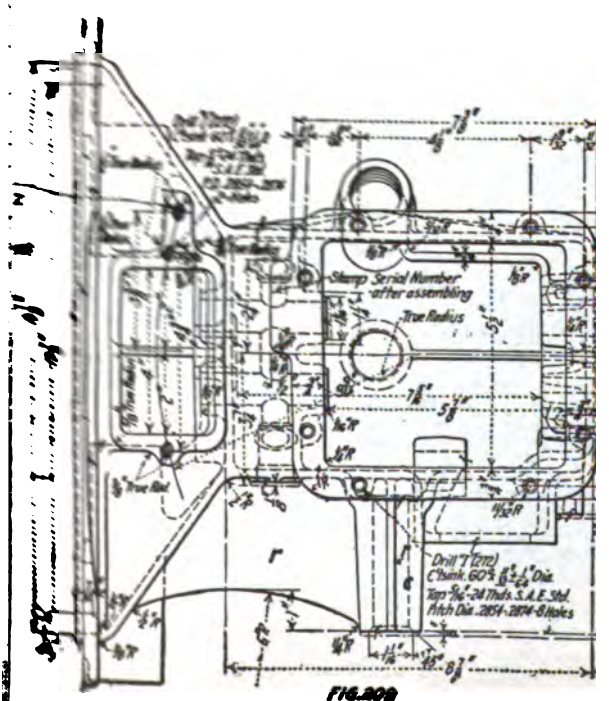
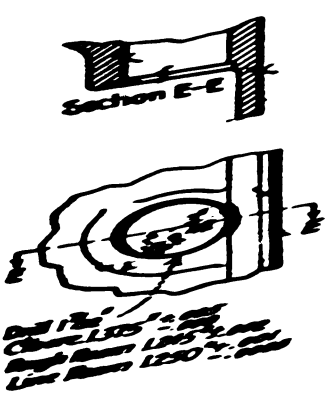

Stamp Serial Number after assembling
True Radius
Drill 7 (.201)
C'bore 60° Ø 1 1/4" Dia.
Tap 3/4"-24 Thds. S.A.E. Std.
Pitch Dia. .2854-.2874-8 Holes

FIG.209



(Facing Pag

With the three-point suspension, sometimes the front is pivotally attached to allow a certain amount of lateral rotative movement to compensate for frame distortions (see Class B case). The rear may be arranged to pivot slightly so that the front end can rise or drop, to accommodate itself to frame flexures. When sub-frames are used, the supporting arms can be made shorter and rest directly on the frame, and twists or flexures in the side members are not directly transferred to the gear case supports; sub-frames, however, mean extra weight and cost.

In the Class B transmission case (Fig. 126) the cover and all the flanges and bearing caps are attached to the case by $\frac{3}{8}$-in. screws. The size of the case is one reason for the comparatively large screws, another is that provision must be made for the class of men who are liable to handle wrenches on the screws and bolts. On the Packard gear case all the screws are $\frac{5}{16}$ in. while in some of the smaller transmissions, they are only $\frac{1}{4}$ in. A small screw is liable to be twisted off when tightened.

In the Class B case, the wall thickness is $\frac{1}{4}$ in., the end walls which support the bearings are $\frac{5}{16}$ in.; they are strengthened in addition by ribs $\frac{3}{8}$ in. thick. Note how well the sections are ribbed which support said bearings and by which the case is suspended.

Figure 208 is a side elevation of the Packard single six transmission case. Figure 209 a plan view with the cover removed. In this gear case the wall is $\frac{3}{16}$ in., except on the top where it is $\frac{1}{4}$ in.; the length of the threaded holes is $1\frac{1}{16}$ in. See the provisions at *b*, (Fig. 208), for the threads. In some of the smallest cases, the wall, where it simply serves as a protection for the parts enclosed and does not carry any of the parts direct, is sometimes only $\frac{5}{32}$ in. thick.

In this Packard case the ribs are $\frac{3}{16}$ in. The screws are $\frac{5}{16}$ in., and it is an advantage if all the tapped holes are alike as the same size screw can be employed. Note how the boss *c* is strengthened by ribs; this boss contains the shaft on which the foot levers are riding or oscillating. Figure 210 is a view looking from the clutch end. Figure 211 is a rear end elevation. Figure 212, a section looking through *F-F* of Fig. 208. Figure 213 is a section through the clutch release shaft bosses, *B-B* in Fig. 208.

CHAPTER VII

REAR AXLES

After the power is transmitted through the transmission (the gearset) it has to pass through the final drive, before it is applied to the wheels. In all rear axles at the final drive, a certain gear reduction takes place to reduce the number of revolutions of the rear axles as compared to that of the propeller shaft. This gear reduction in passenger cars varies between about three to one, to six to one, the average being 4.53 (see Statistical Number *Automotive Industries*, 1922).

There are five general types of final drives in use. First: The chain and sprocket type where the transmission case is placed amidship, a jackshaft, forming a part of the transmission, is positioned across the frame and has small sprockets at each end; the wheels are provided with large sprockets, which are driven from the smaller sprockets of the jackshaft by means of chains. The second type of final reduction is the bevel gear drive, having either straight bevels or spiral bevels, as mostly used on passenger cars; the third type is the worm gear drive, which is the most common type used on trucks; the fourth type is the double reduction type of rear axle, where two reductions are made between the propeller shaft and the rear axle; the fifth type (which is also a double reduction type) is the internal gear drive, in which the first reduction is accomplished by means of bevel gears in the center of the axle, and the second by internal gears in drums or rings attached to the wheels; small pinions or spur-gears driving the larger internally cut gears attached to the wheels.

In America practically all passenger cars are driven by bevel gears, while in trucks the worm drive axles preponderate. The proportion in truck models is as follows: (see *Automotive Industries*, Feb. 16, 1922). Worm, 72 per cent, internal gear, 16.9 per cent; chain 3.9 per cent; bevel 5.4 per cent; double reduction 1.8 per cent.

Axle Housing for Passenger Cars.[1]—The load sustaining portion of the rear axles has passed through a number of stages of development. In some of the earlier forms the axle housing was divided through a horizontal center line, the two halves having been bolted together; in some designs, straight tubes were driven into and riveted to a malleable iron center casting which had an opening in the upper half for assembling the differential and the gears. This opening was sealed by a cover fitting over the axle housing and clamped to the main body of the casting along a horizontal center line, see Fig. 214. Fig. 215 shows the side tube members, opening in a funnel shape that fits over and is riveted to a corresponding surface of the center

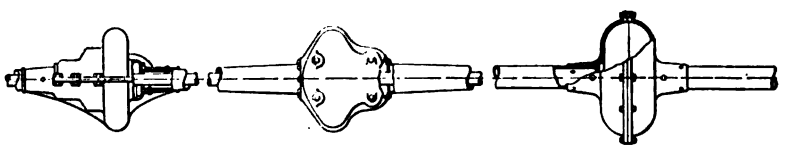


FIG. 214. FIG. 215. FIG. 216.

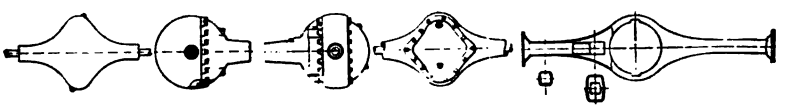


FIG. 217. FIG. 218. FIG. 219.

FIGS. 214 to 219.—Rear axle housings.

casting. The cover is resting on a flat seat, which is far enough above the center line of the axle to permit the full engagement of the funnel shaped tube ends with the center casting. In this construction a great deal of strain is put on the rivets. In Fig. 216 the center casting is divided through a vertical center line and tubes are driven in and riveted to the sides of the center housing.

Pressed Steel Housings.—Figure 217 shows one of the early forms of pressed steel housings consisting of two central stampings welded through a vertical center line. The tubes at the end were welded again to the center, thus making three welded joints in vertical lines. The stamping is closed at the rear and the differential and driving gears are self contained in a malleable iron casting or carrier which is bolted along a vertical line to a turned over flange on the steel stamping.

[1] See J. G. PERRIN, Evolution of the Rear Axle, *S.A.E. Transactions*, 1916, Part 2.

Figure 218 shows a more accessible construction, as used by a large number of car makers. The carrier casting is bolted to the front of this stamped housing, and the cover, of either cast or pressed steel, is bolted to the rear, as seen. This type can easily be constructed in various styles of stampings and thicknesses of metal. Most of these housings are stamped from a good grade of carbon steel suitable for press work. They are made as a rule of two stampings, symmetrical from a horizontal or vertical center line.

Figure 219 is a pressed steel housing in which a square tubular section is used under the spring seat and the ends are flared up for the brake spider anchorage. This form gives a secure spring seat anchorage for the Hotchkiss drive (see chapter on Springs).

In the larger housings, for cars weighing about 4,000 lb., the material used is $\frac{3}{16}$ in. thick and the butt joint can then be successfully welded on a horizontal line. These housings range in weight from 30½ lb. for those made of $\frac{1}{8}$ in. thickness of metal for light cars, in which the springs are supported at the extreme ends of the housing, to 55½ lb. for designs made of ¼-in. metal, for cars weighing 5,000 lb. Those for cars weighing 4,000 lb. made of $\frac{3}{16}$-in. metal and including the sleeves weigh about 54 lb. The arch in the center of the housing must have an opening large enough to permit the assembly of the bevel gears, the size of the latter being determined by the power to be transmitted and the required gear reduction. The load carrying capacity of the housing is determined by the diameter of the ends of the housing. These diameters generally range from 2½ by $\frac{3}{16}$ in. for cars weighing 2,000 lb. (without passengers) to 3 × $\frac{3}{16}$ in. for those weighing 4,000 lb. and 3½ × ¼ in. for 5,000 lb. cars.

Bevel Gear Drive.—Formerly a straight tooth was the prevailing form in bevel drive gears. The straight-tooth bevel gears are inclined to be noisy unless the fit and the adjustments are perfect. With the improvements made in engines and transmissions, these units operated more quietly and the difficulties became greater in furnishing straight-tooth bevel gears as noiseless as were demanded. This condition was aggravated by quantity production, which made it important that the gears operate as noiselessly as possible with a minimum of adjustments, fitting and trying, after a chassis was put on the road. Furthermore, the gear reductions, on account of greater range of speed on the direct drive, necessitated either smaller pinions or larger driving

gears. These too, added to the difficulties of getting quiet gears. A solution of this difficulty was sought and as a result the spiral bevel gear was developed and is used to-day on about 90 per cent of the number of passenger car models on the market in this country.

On high price as well as low price cars, the necessity of a small pinion, to obtain a large gear reduction, has brought about the integral construction in which the pinion and the pinion-shaft are made in one piece, as is now the common practice. The spiral bevel operates satisfactorily with ten-tooth pinions; some manufacturers use as few as seven or eight teeth or even six, and in only a few cases, in the larger cars, has it been necessary to make the main driving bevel so large that more than 11 teeth are needed in the pinion. The use of the spiral bevels has resulted in several detail changes in the mounting of the pinion-shafts and differentials, as the tendency for the pinion to screw in or out of the main drive bevel gear necessitates better provisions for end-thrust in both directions, especially on the pinion-shaft bearings. Bearings capable of satisfactorily taking end-thrust as well as radial lead have solved this problem most satisfactorily.

The use of a pinion-shaft which extends forward, surrounded by a tube fastened rigidly to the axle housing and supported at the front end so as to take the torque and sometimes the drive also, which formerly was the common practice, has largely been superseded by the short pinion-shaft and two universal-joints on the propeller-shaft. In the latter construction the torque is taken through arms fastened to the axle housing, rigidly, or else, through the medium of the side springs, the latter having seats fastened rigidly to the housing. This construction of driving through the springs is commonly called the Hotchkiss drive, from the name of the Hotchkiss Company of France, which first introduced it, and its use is increasing (see chapter on Springs).

Methods of Supporting Pinions.—Three general forms are used to support the pinion-shaft; one, in which the pinion is supported between the bearings (see Figs. 273 and 275) another in which the pinion has a bearing directly at its back, taking both radial and thrust loads and with the other bearing at the extreme end of the concentric tube or torque tube, (Fig. 220) (see description on page 200). This form gives a wide separation of bearings, but is not so much in vogue to-day as formerly. The third and most

common form is that in which one bearing is immediately in front of the pinion (toward the transmission) and the other bearing a short distance forward, (see Figs. 221 and 222), the distance being determined by the proper distribution of load on the two bearings. In fact, the length of the pinion-shaft is determined by the proper spacing of bearings to get a proper distribution of the load. The last two figures give the most usual forms of pinion-shaft mountings.

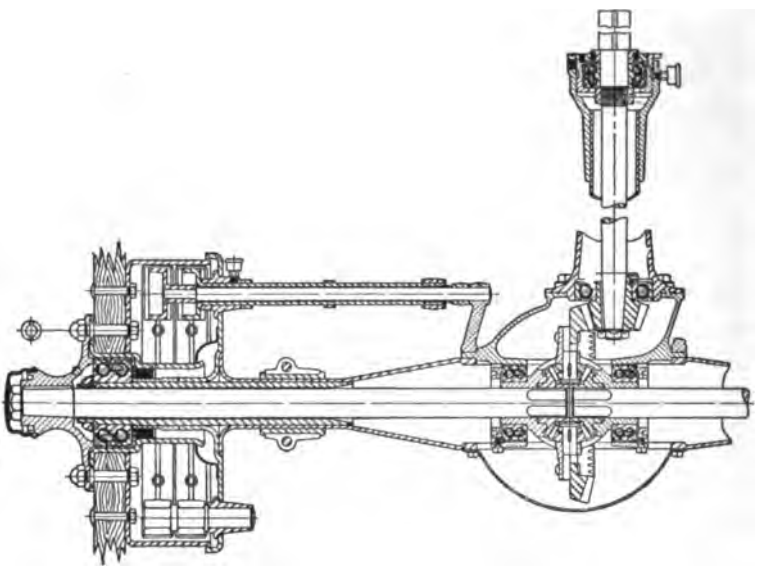


FIG. 220.—Rear axle with concentric torque tube.

The pinion is made integral with the shaft (Fig. 221) and the double row ball bearing is at the front end of the shaft. Adjustment of the pinion is possible in these constructions with the view of securing accurate meshing of the gears. In Fig. 221 the bearing nearest the pinion is single row, and as the outer race is not restrained from endwise movement it takes only radial load. The end thrust is transmitted by the pinion shaft to the double row bearing which is mounted in an adjustable housing at the front end and which can be screwed in or out of the pinion gear-shaft carrier. The double row bearing has clamped inner and outer races and provides for both thrust and radial loads. After the proper adjustment is made, the split front end of the housing is clamped by a through bolt at *b* to retain the adjustment.

Figure 222 is more costly but is superior in providing a more positive locking means and absolute alignments of bearings. The bearings are assembled in a carrier member, threaded at the upper end that it may be screwed back and forth in the main casting. This bearing carrier is provided with a series of longitudinal slots evenly spaced about its circumference at the center, and a locking plate *c* with a tongue projecting from its inner face is designed to fit in any one of the slots and thus supplement the

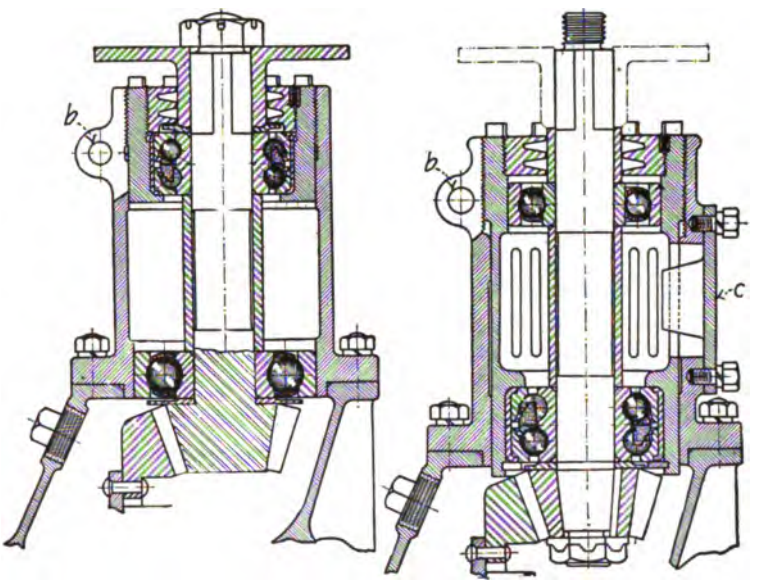


FIG. 221. FIG. 222.

FIGS. 221 and 222.—Pinion-shaft mountings.

adjustment lock provided by clamping bolt. To vary the pinion position, it is necessary to remove the locking plate as well as releasing the clamp bolt at *b*. The bearing carrier can then be turned as desired, though care is needed to have one of the slots always in such a position that the tongue of the locking plate will register when it is replaced after adjustment of the pinion position is completed.

The use of a small number of teeth in the pinion has made the integral pinion-shaft and pinion almost a necessity. When the pinion is made separate, it is generally secured to the shaft by a taper-fit, key and nut. The universal-joint flange, or coupling, is generally fastened to the front end of the pinion-shaft in the

same manner, both tapers being S.A.E. standard. Other methods of securing the coupling flange to the pinion-shaft are by means of square or spline fittings.

Main-drive Bevel Gear.—The bevel ring gear is fastened to the differential flange by rivets or bolts (see Figs. 251 and 272). Satisfactory results are obtained when the riveting is done cold, either by hand or by a power press; the latter method having been developed to a point where it is reliable in seating the driving bevel gear uniformly against the flange, to give a true running condition.

Forms of Differentials.—The differential itself has remained fundamentally the same for several years, being constituted of a nest of either spur or bevel gears; the bevel type predominating. The usual form of bevel-gear differental consists of a spider having from two to four trunnions, these being clamped at their ends between the two halves of the differential casing. The pinions revolve on these trunnions which form a bearing surface of hardened steel. Sometimes a bronze bushing is inserted in the pinion, but in the majority of cases no extra bushings are needed, since the hardened pinion runs well on the hardened spider pins.

A differential developed to meet cost reduction has a one-piece casing with two pinions, (see Fig. 228). The case is a one-piece malleable casting. Some little juggling is necessary in laying out some of these differentials to get compactness and ease of assembling. It is necessary to assemble it with the main bevel gear riveted in place on the differential-case flange, as the teeth of this gear usually overhang the differential side-gear.

The positive-drive form of differential, in which the drive is taken on the wheel which gives the best traction, has been developed commercially in several forms. The advantage of this construction is due to the increased friction of the differential pinions.

Axle Shafts.—Axle shafts can be classified in general as the full-floating or fixed-hub types. In the full-floating type, (Figs. 289 and 345) all the bending stresses due to the load and skidding force are carried by the housing, while the axle shaft or driving shaft turns freely in the housing and transmits only the driving torque. The inner end of the shaft floats in, and is driven by, the differential side-gear, by either splines or by a square fitting. The outer end of the shaft drives the wheel hub either through a flange

secured to the axle shaft by keys, splines, or by a square fitting. The outer end of the shaft drives the wheel hub, either through a flange formed integrally with the shaft, the periphery of the flange having teeth that engage with corresponding teeth on the wheel hub, or by a flange secured to the axle shaft by keys, splines or square fittings. The wheel hub is supported independently on the housing by a pair of bearings spaced a certain distance apart. The greater this distance, the smaller is the bearing pressure when turning corners or side skidding, or when the wheel strikes the curb, and this may impose a much greater stress on the bearings than the radial load (the vertical load) due to the weight of the car or truck.

The fixed-hub type takes two general forms, (the semi-floating, Figs. 275 and 321 and the three-quarter floating Figs. 225 and 246), both having the wheels attached rigidly to the driving-shafts. In the semi-floating type, the inner end of the shaft floats in the differential as in the full-floating type; the outer end has a single bearing fitted over the ends of the shaft and inside the tube, which forms the end of the axle housing.

While a number of concerns use the full-floating type, it is claimed by its opponents that in case of skidding into curb stones the whole axle housing may be damaged, and that this axle is more expensive and heavier than the semi-floating type. The full-floating axle has also a larger number of bearings. The advocates of the full-floating type on the other hand claim it a great advantage to separate the member which transmits the torque from the load-carrying member.

In the semi-floating type, the axle housing as well as the drive shaft carry the entire radial load as well as the stresses due to skidding or when turning corners; in addition the shaft transmist the power to the wheels as in the previous type. In practice it is found that serious trouble is rarely encountered from the ordinary radial load, but when either wheel strikes an obstruction a glancing blow or brushes against the curb, or skids, the bearing pressures are tremendously increased; in the full-floating type this bearing pressure is ordinarily much higher than in the semi-floating type, for in the latter, the two bearings (one at the wheel, the other in the differential) are much further apart.

It is further claimed for the semi-floating axle, that if the pressure against the side of the wheel is so severe as to cause damage it can only bend the driving axle, the replacement of

which is more quickly accomplished and at lesser cost than the damage that would result in a full-floating axle, in which case the axle tube on which the bearings are supported are liable to be damaged and this may involve the expense of a new axle housing. Manufacturers of the full-floating axle, on the other hand, claim that the axle tube which supports the bearing can be made stronger than the shaft, and that the drive shaft can be made lighter than in the semi-floating type.

Another form of the fixed hub type is the commonly called three-quarter floating type, (Fig. 220). The inner end of the shaft floats in the differential, while the outer end is fixed to the wheel hub (it is not in contact with, or touches, the axle housing, there being a clearance between the shaft and the axle housing tube). The inner race of the wheel bearing (this bearing is in practice a single or double row, ball or roller type), is supported on the axle housing tube, *i.e.*, on a tubular extension of the housing, and the hub revolves on the outer race of this bearing which supports the radial load on the wheel. The inner race is clamped to the axle housing tube, while the outer race is clamped to the wheel hub, thereby keeping the wheel firmly in place. New Departure double row ball bearings are used in this example, which take the thrust in both directions. The wheel driving shaft (the live axle) transmits the driving torque and as it is rigidly attached to the wheel hub outer flange, it serves as a pilot member which keeps the wheel in alignment when skidding or when it is subjected to lateral stresses; in other words a portion of the load, arising from a skid, is carried back to the differential bearing through the shaft.

In this construction, as well as in the semi-floating type, the axle drive shaft must be rigidly secured to the wheel hub, and there must be a good running fit in the interior of the differential case hub.

The fixed hub type of construction, (both the semi-floating and the three-quarter floating), is better able to withstand skids or side thrusts between wheel and road surface than the full-floating axle. On the other hand, the radial load, the direct load on the wheel due to the load resting on the springs, is supported by one bearing only—the wheel bearing— while in the full-floating type both bearings support the radial load. (See Loads on wheels and stresses in shafts,Chapt. XI.)

All three types of axles are in common use on passenger cars,

and give satisfaction when properly designed and constructed. About 30 per cent of the number of passenger car models use the full-floating type, while the semi-floating and the three-quarter floating type each, are used on slightly more than one third of the number of models.

Brakes.—(See also Chapt. XI.) The brakes used on rear axles are either of the contracting or the expanding type. Contracting brakes are generally applied to the same drums on which the internal expanding brakes operate.

The contracting brake band in common use, consists of a band of steel, with a lining riveted thereto, containing woven asbestos fiber. This steel band is .019 in. thick, or thicker and varies in width on passenger cars from about 1¾ to 2½ in. The lining material is riveted to the steel band as a rule by copper rivets.

The brake band, as a whole, is generally anchored at one side and the contracting levers at the opposite side. Contracting brakes are sometimes made with forged or cast bands or shoes, hinged at one or two points where they are anchored. The lever action is almost invariably of the double-acting type, as seen in the illustrations.

A greater variety of designs is used in bringing the band in contact with the brake-drum in the internal brake than in the contracting or external type. The simplest and most common form of internal brake is operated by a cam. The brake band, or shoe, is anchored at one point and the ends of the band on the opposite side are fitted with brackets between which the operating cam turns, thereby opening or expanding the brake band against the drum. A strong contracting spring is used to hold both the ends of the internal band or the shoes in contact with the cam and to hold them out of contact with the drum. The band is made of a piece of heavy band steel, which is generally continuous, while when shoes are employed they are forgings or castings, hinged at the point of anchorage to the brake spider. When a band is used in the internal brake, anchored in the middle, the two halves will tend to wrap themselves into contact with the drum, and this is called the wrapped type of internal brake. When the band is not anchored but is free to wrap its entire length against the drum it is called the full wrapping type (see Figs. 324 and 339).

In nearly all designs of passenger cars, the same lining material is used for both, the internal and the contracting brakes. With

the cam brake provision should be made for getting a proper length of lever on the end of the shaft which operates the expanding cam. With a properly designed cam and an adequate length of lever the cam brake operates very successfully and is simple in construction.

An internal brake operated by toggle joints, is very effective, but has more parts and is more costly than the cam brake. (See Figs. 224 and 225.)

Generally speaking, foreign practice has always favored the transmission or propeller-shaft brake for service work and a rear wheel brake for emergency work. American car builders, as a rule install both service and emergency brakes on the rear wheels, although some use one brake on the rear axle and one between the transmission and the rear axle, this being called the propeller shaft brake. This type of brake is effective with a small pedal pressure, but designers must consider the additional strain on the propeller-shaft joints, bearings, driving gears and shafts. The placing of both brakes on the rear wheels is a great advantage from the car assembler's viewpoint.

Example of Passenger Car Rear Axles.—Figures 223 and 224 are assembly views of the rear axle of the Oldsmobile, Model 43A, four-cylinder, having a bore and stroke of $3\frac{1}{16}$- × $5\frac{1}{4}$-in. piston displacement 224.3 cu. in. The wheelbase of this car is 115 in. and its weight complete (without passengers) about 2,900 lb. The ratio in the rear axle is 4.666 to 1. In this axle the housing is split in the center vertically and bolted together; a $\frac{1}{2}$-in. truss rod being provided underneath for additional strength. (For calculations of truss rod stresses see Chapt. X.) The tube of the housing (axle tube) is $2\frac{1}{2}$ in. outside diameter and $\frac{7}{32}$ in. thick. In this axle the torque reaction is taken through a torque tube rigidly attached to the rear axle housing, and extending to a ball joint to the rear end of the transmission. The drive pinion is keyed to the taper end of the long propeller-shaft. Note at *A*, how a flange of the pinion forms the lower seat for the ball race. By this means a longer key-and-taper fitting is established between the shaft and the pinion. The axle is three-quarter floating. The wheels and the drive shafts are supported on Hyatt roller bearings; side pressure against the wheels in either direction, is resisted by the thrust ball bearings *C* at the differential. If the lateral stress is applied from the outside (for instance, when hitting the curb) the thrust

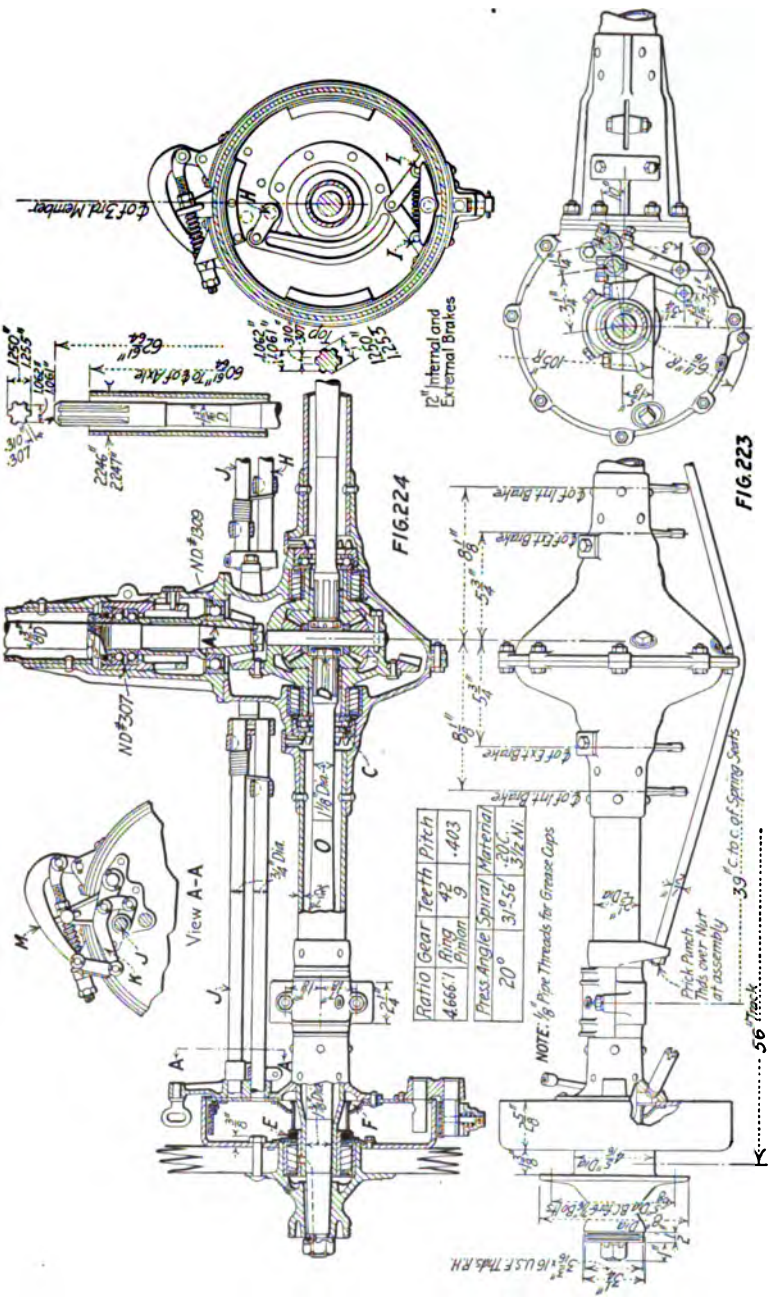


FIG. 224
FIG. 223
12" Internal and External Brakes
View A-A
Ratio Gear Teeth Pitch
4.666:1 Ring 42 .403
Pinion 9
Press Angle Spiral Material
20° 31°-56' .20C. 3½ Ni.
NOTE: ⅛" Pipe Threads for Grease Caps
Pitch Punch Nuts over Nut at assembly
39" C. to C. of Spring Seats
56" Track

FIGS. 223 and 224.—Rear axle of Oldsmobile four-cylinder car.

is transmitted longitudinally through the shaft *O*, through the differential gears and its housing to the ball thrust bearing on the opposite side. A stress tending to pull the wheel away from the car is resisted by the thrust bearing of the same side. To enable it to do so, however, the drive shafts must be rigidly attached to the differential side gears; this is accomplished in this design by a shoulder on the spline end of the shaft, the side gear being drawn up tight and locked in place by the nut *D*. To prevent oil leakage from the axle housing, a felt ring is placed at *E*, and any oil that should find its way past this packing is collected in *F*. The diameter of the propeller shaft is $1\frac{3}{16}$ in. at its weakest point, while the axle shaft is $1\frac{1}{8}$ in. The object of making the propeller-shaft (which in this case also forms the pinion-shaft) of larger diameter than the axle shaft, is due to its long length, over 60 in., which renders it liable to whip, or whirl (see page 123) hence a greater dimension is desirable even though the torque is smaller, (but its speed is higher). The internal expanding brake is toggle operated, and the brake shafts are $\frac{3}{4}$ in. in diameter as shown.

The operation of the internal brake may be seen from the right hand top view. As the brake shaft, *H*, is moved counter-clockwise, the long curved lever spreads the ends of the toggle link *I*, thereby expanding the brake band. The external brake is seen from view *A-A*: Moving the brake shaft *J* counter-clockwise, lifts the lever *K*, which is pivotted at its right end, and this forces the left end of the brake band lever *M* upward, thereby contracting both ends of the brake band.

Figure 225 are assembly views of the rear axle (Weston Mott type) used on the Oakland car, having a six-cylinder engine with a bore and stroke of $2\frac{13}{16} \times 4\frac{3}{4}$ in., piston displacement 177 cu. in. The axle is of the three-quarter floating type and the wheelbase is the same as that of the preceding car, 115 in., but the engine is of smaller capacity. The drive is taken through the springs (see chapter on Springs, Hotchkiss drive) instead of through a torque tube as in the preceding case. The axle is somewhat lighter, being used on a lighter car; the diameter of the axle tube is $2\frac{1}{4}$ in. with a wall thickness of $\frac{3}{16}$ in. The axle shaft diameter is $1\frac{1}{16}$ in. The truss rod, $\frac{1}{2}$ in. in diameter, is attached to the brake supports, thereby strengthening the axle tube ends and counteracting the tendency of the wheels to toe outward at the bottom.

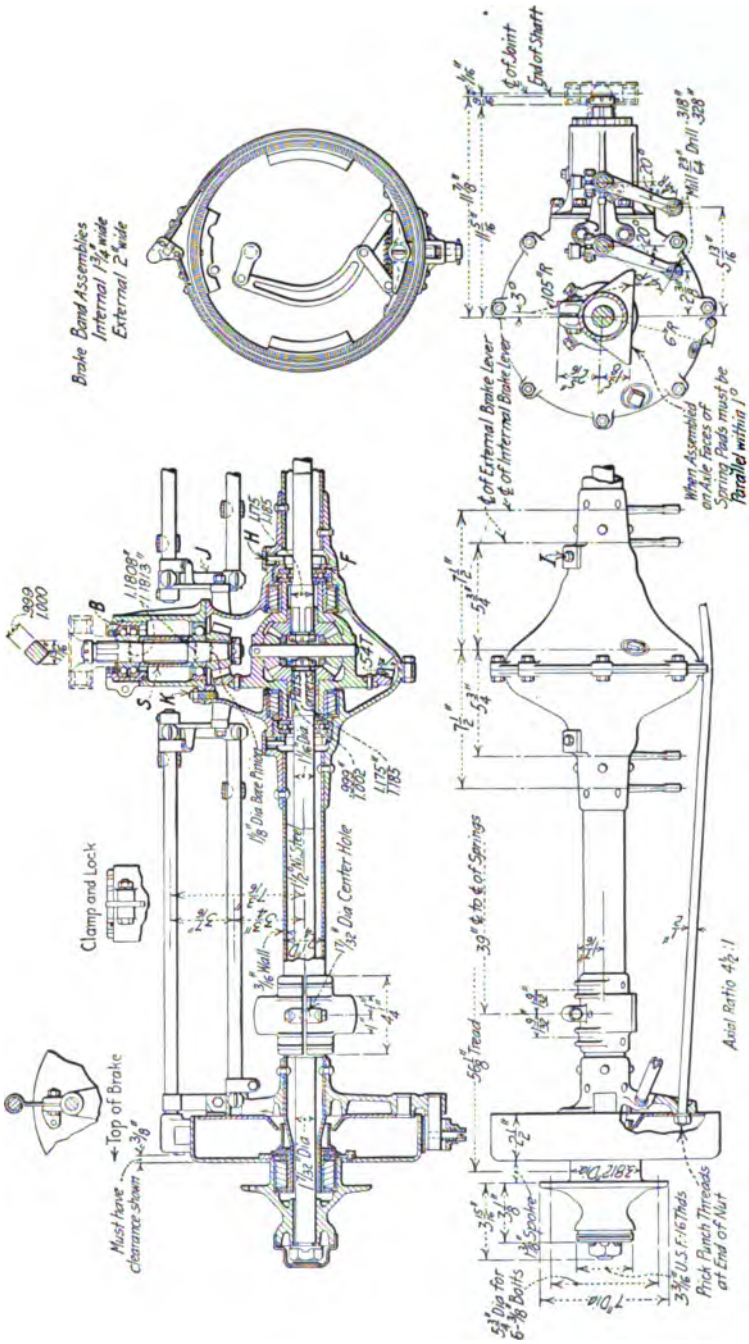

Brake Band Assemblies
Internal 1¾" wide
External 2" wide
Clamp and Lock
Top of Brake
Must have clearance shown
Axial Ratio 4½:1
℄ of Joint End of Shaft
℄ of External Brake Lever
℄ of Internal Brake Lever
When Assembled on Axle faces of Spring Pads must be Parallel within .10
℄ to ℄ of Springs
Prick Punch Threads at End of Nut

FIG. 225.—Rear axle of Oakland car (Weston Mott type).

Figure 226 is a detail of one-half (right-hand) of the center portion of the axle housing called the bevel gear housing. It is a malleable casting, the axle tube being inserted in the hub of the housing and riveted thereto, as seen from the assembly. The two halves of the housing are bolted together by seven, $\frac{5}{16}$-in. bolts.

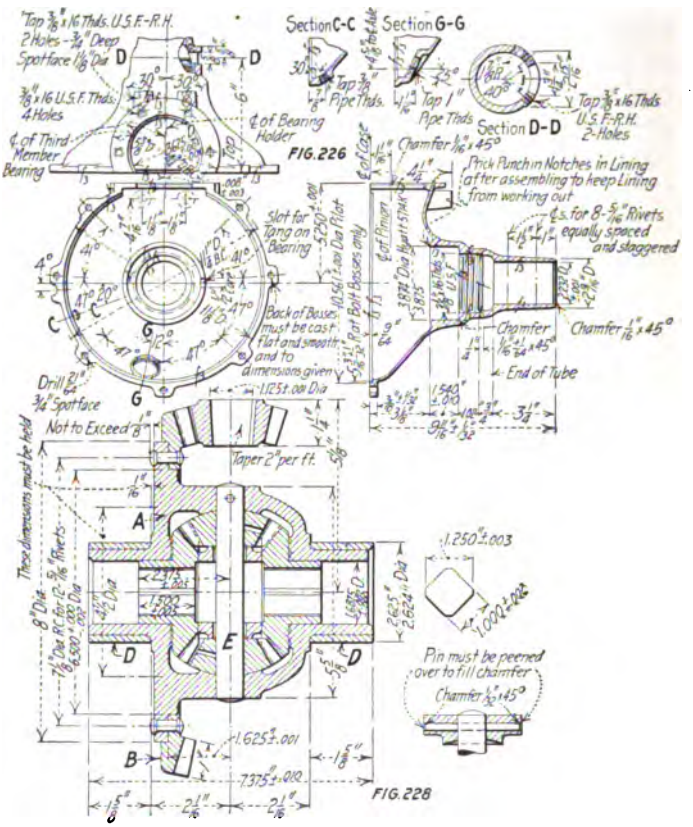

Section C-C
Section G-G
Section D-D
FIG. 226
FIG. 228
Chamfer 1/16 x 45°
End of Tube
Taper 2" per ft.
Pin must be peened over to fill chamfer
Chamfer 1/32 x 45°
Back of Bosses must be cast flat and smooth and to dimensions given
Slot for Tang on Bearing
Prick Punch in Notches in Lining after assembling to keep Lining from working out
Not to Exceed 1/8"
These dimensions must be held

FIG. 226.—Bevel gear housing (right half). FIG. 228.—Differential (two-pinion type) (Oakland rear axle).

The wheels and the differential case are supported on Hyatt roller bearings; ball thrust bearings are provided at each side of the case to take the lateral stress against the wheels, as in the preceding design; the thrust bearing on the left also taking the thrust due to the bevel drive gears. The pinion shaft is supported on ball bearings, the rear bearing being free to float while

the double row bearing in the front has its upper and lower races clamped to take thrust in both directions.

Figure 227 is a detail of the drive pinion adjusting sleeve, sometimes called the pinion flange. It is a malleable casting having a wall thickness in the center of $\frac{3}{16}$ in. Note how the six ribs are placed in order to increase its strength without materially increasing the weight. This sleeve is attached to the bevel gear housing, by means of six, $\frac{3}{8}$-in. studs, see *K* in assembly. After the sleeve

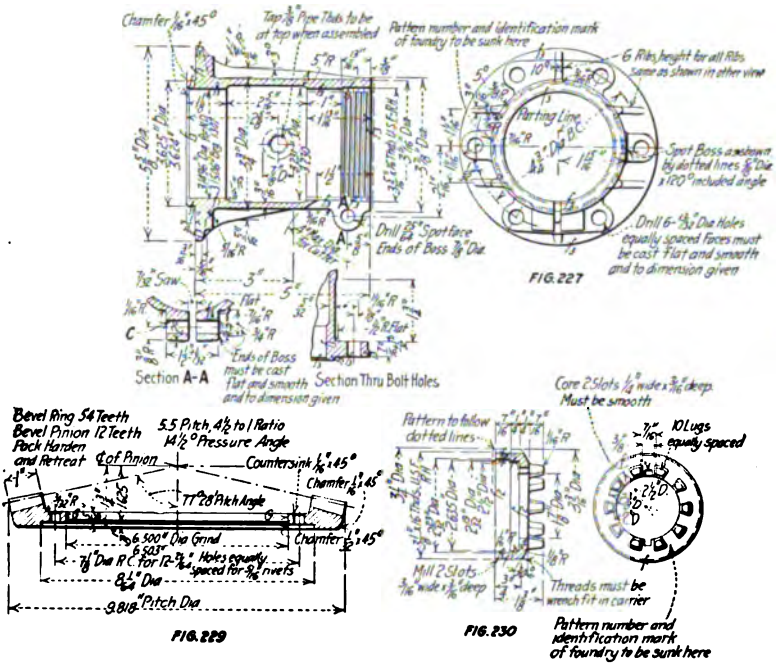

Chamfer 1/16" x 45°
Tap 3/8" Pipe Thds to be at top when assembled
Pattern number and identification mark of foundry to be sunk here
6 Ribs, height for all Ribs same as shown in other view
Parting Line
Spot Boss as shown by dotted lines 3/8" Dia x 120° included angle
Drill 6-13/32" Dia Holes equally spaced Faces must be cast Flat and smooth and to dimension given
FIG. 227
Drill 25/64" Spot face Ends of Boss 3/8" Dia
3/16" Saw
Flat
Ends of Boss must be cast Flat and smooth and to dimension given
Section A-A
Section Thru Bolt Holes
Bevel Ring 54 Teeth
Bevel Pinion 12 Teeth
Pack Harden and Retreat
5.5 P. Pitch, 4 1/2 to 1 Ratio
14 1/2° Pressure Angle
℄ of Pinion
Countersink 1/16" x 45°
Chamfer 1/16" x 45°
77° 28' Pitch Angle
6.503" Dia Grind
Chamfer 1/16" x 45°
7/8" Dia P.C. for 12-7/16" Holes equally spaced for 7/16" rivets
8 1/4" Dia
9.818" Pitch Dia
FIG. 229
Pattern to follow dotted lines
Core 2 Slots 1/4" wide x 3/16" deep Must be smooth
10 Lugs equally spaced
Mill 2 Slots 1/16" wide x 1/8" deep
Threads must be wrench fit in carrier
FIG. 230
Pattern number and identification mark of foundry to be sunk here

Fig. 227.—Pinion adjusting sleeve. Fig. 229.—Ring gear. Fig. 230.—Bearing adjuster (Oakland rear axle).

and adjusting collar *B* are in place, a bolt, inserted at *C* through the lugs of the slotted end, locks this collar against displacement. The bevel pinion spacer *S* (Fig. 225) is a gray iron casting.

Figure 228 is an assembly view of the differential which is of the two pinion type. As seen from this figure, the differential case *A* is a one-piece malleable iron casting. Instead of a differential spider as shown in Figs. 251 and 325 one shaft *E* of cold rolled steel, pinned in place, forms the bearing for both the differential pinions. This differential forms a simple, compact and inexpen-

sive unit. *D-D* are bearing sleeves made of high carbon steel tubing. At *B* is the drive gear or ring gear (see Fig. 229) which is rivetted to the differential case by 12-$\frac{5}{16}$-in. rivets. This ring gear is made of low carbon steel, carbonized and heat treated. The pinion is made of nickel-alloy steel. The ratio of the standard gears is 4½ to 1. All the details as to pitch, number of teeth, etc., can be seen from this drawing. The two bevel pinions of the differential have 12 teeth of 5 pitch; the two bevel side gears have 18 teeth.

Figure 230 is the malleable iron differential bearing adjuster, marked *F* in the assembly; two of these are used, one on each side of the differential. Through them the differential case may be moved to the right or the left to obtain the correct position for the meshing of the bevel gears, and the thrust bearings may be given the desired amount of play. Adjustments can be made from the outside of the housing by turning the adjusters to the right or left by their prongs; afterwards they are locked by the differential adjuster lock *H*, which is bolted to the housing by two cap screws *I*.

Figure 231 is a detail of the axle shaft or main shaft, made of chrome-nickel steel, heat treated. It has a taper-fit, key and nut, for attachment to the wheel, and a square-fit and nut for the differential side gear.

Figure 232 is a detail of the malleable iron brake support or brake spider. At *T* is the truss rod anchorage; *V* and *W* are the anchorages for the external and internal brake bands respectively. The brake drum is 12 in. in diameter, and the width of the external brake band 2-in. and that of the internal band 1¾ in. The internal brake is toggle operated, while the external brake is of the contracting band type, of extremely simple design, as can be seen from the assembly.

Figure 233 is a detail of the brake lever shaft support, marked *J* in the assembly views. It is a malleable casting, and is bolted to the bevel-gear housing as shown.

The axle housing of the Columbia rear axles is of tubular section, made from one piece of steel and having a single welded seam. Figure 234 shows the one-piece housing semi-finished and Fig. 235 the housing complete. The Columbia passenger car axles are of the three-quarter floating type. The differential and gears are carried in the differential carrier *C* (Fig. 236) bolted to the front side of the housing; this carrier in conjunction

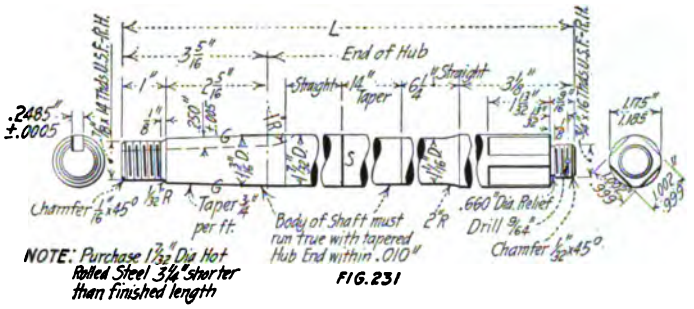

L
End of Hub
Straight
Taper
Straight
.2485" ±.0005
Chamfer 1/16" x 45°
1/32" R
Taper 3/4" per ft.
Body of Shaft must run true with tapered Hub End within .010"
2" R
.660" Dia. Relief Drill 9/64"
Chamfer 1/32" x 45°
NOTE: Purchase 1 3/32" Dia Hot Rolled Steel 3 1/4" shorter than finished length

FIG. 231

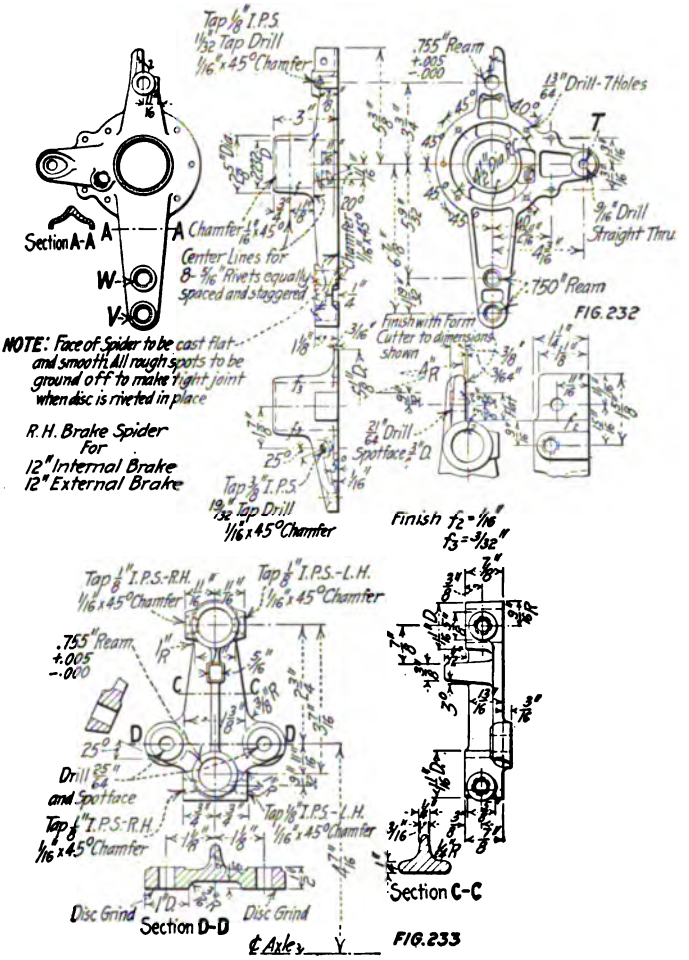
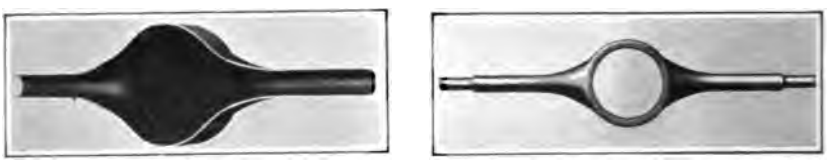

Tap 1/8" I.P.S.
11/32" Tap Drill
1/16" x 45° Chamfer
.755" Ream +.005 -.000
13/64" Drill-7 Holes
Section A-A
Chamfer 1/16" x 45°
Center Lines for 8-5/16" Rivets equally spaced and staggered
9/16" Drill Straight Thru
.750" Ream
NOTE: Face of Spider to be cast flat and smooth. All rough spots to be ground off to make tight joint when disc is riveted in place
Finish with Form Cutter to dimensions shown
R.H. Brake Spider For 12" Internal Brake 12" External Brake
21/64" Drill Spotface 3" D
Tap 3/8" I.P.S. 19/32" Tap Drill 1/16" x 45° Chamfer
Finish f2 = 1/16" f3 = 3/32"
Tap 1/8" I.P.S.-R.H. 1/16" x 45° Chamfer
Tap 1/8" I.P.S.-L.H. 1/16" x 45° Chamfer
.755" Ream +.005 -.000
Drill 25/64" and Spotface
Section C-C
Disc Grind
Section D-D
Disc Grind
℄ Axle

FIG. 232

FIG. 233

FIG. 231.—Axle shaft. FIG. 232.—Brake support. FIG. 233.—Brake lever shaft support (Oakland rear axle).

with the housing and the housing cover, bolted to the back, constitute the receptacle for holding the lubricant. The oil being maintained to a level, even with the bottom of the oil level plug in the cover. It is recommended that a semi-fluid oil, such as steam cylinder oil, be used, which is poured into the housing



FIGS. 234 and 235.—Columbia axle housing.

through the filler plug on the top of the malleable carrier on the front of the axle. This drawing shows the Columbia Axle Co.'s model 50,000, which is designed for passenger cars weighing from 3,200 lb. to 4,000 lb. without passengers. The seamless steel tube *t* strengthens the axle housing and forms the support for the wheel bearings. The steel housing is made of

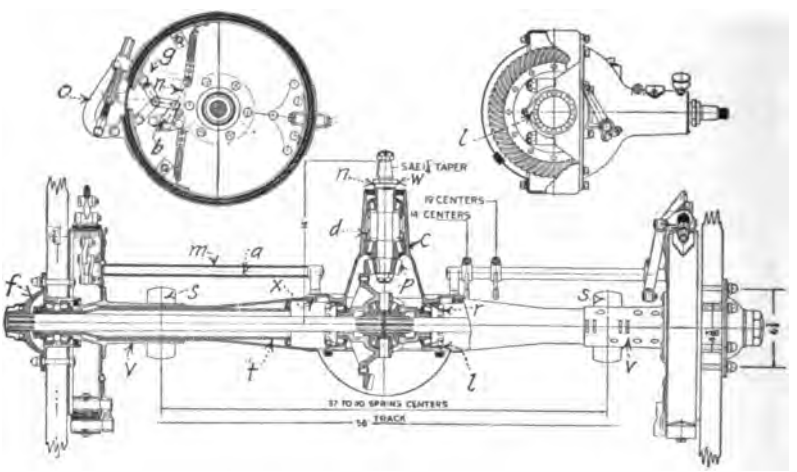


FIG. 236.—Rear axle of Columbia Axle Co.

$\frac{3}{16}$-in. stock. At the front face, at *X* the metal is doubled over to strengthen it where the differential carrier is bolted to it. This carrier, *c*, is a one-piece malleable iron casting and supports all the gears and bearings as shown. It is threaded on a portion of its bore for the adjusting collar *d* which serves to hold the bearings in place and take the thrust in both directions. Adjust-

ments to the gears may be made through an aperture from the outside of the carrier, through which the adjuster can be turned to the right or the left by slots provided for this purpose; afterwards it is locked in position as that of (Fig. 222).

Figure 237 is a detail of the pinion, which, as seen, is now made integral with the pinion-shaft, instead of fastened to it by means of a taper fitting and key, as shown in the assembly at *p*. They are made of nickel steel S.A.E. 2320, the wearing surfaces being hardened. The pinion has eight teeth, while the drive gear has 39, (see Fig. 238). The gears are spiral bevels having a pitch of 3.5 and a 20° pressure angle. The various dimensions of the gears may be seen from the drawings. The drive gear is riveted to the differential housing by 12-$\frac{5}{16}$-in. rivets.

All the bearings are of the Bock tapered roller type. The gears are matched at the factory, to secure quiet operation and are adjusted under load by means of threaded collars *r*; the latter permits the adjustment of the bearings and also moving the entire differential housing to the right or the left in order to adjust the backlash between gear and pinion. Said collars are then locked in position by locking plates *l* which in turn are fastened to the carrier by a screw.

The pinion bearing can be adjusted by nuts *n*, which are locked by lock washer *w*. Figure 239 is a detail of the drive shaft, (made of S.A.E. 2340 steel) which extends from the differential to the driving flanges of the wheel. The ends of the shaft have 10 splines where they fit into the differential case, forming a sliding fit, and 10 splines to form a press fit in the wheel driving flange, as shown dotted in this figure and marked *f* in the assembly. Figure 240 is a detail drawing of this drive flange, made of S.A.E. 1035 steel, drop forged. Its brinell hardness is made 200 to 250. It may be noted that metal thickness increases as the diameter decreases, since the stress is greatest near the center. Six special bolts passing through the wheel hub, wheel, and brake drum attach these parts together.

The brake drum is made of $\frac{3}{16}$-in. stock, a steel stamping; it is 16 in. in diameter and 3 in. wide, with a tolerance of +0, $-\frac{1}{32}$ in. By removing the drive flange bolts, the drive flanges and shafts may be removed from the axle housing without jacking the wheel off the ground. This is one of the advantages claimed of this construction over the fixed hub or semi-floating type of axle. Both brakes are of the wrap-up types; the external brake

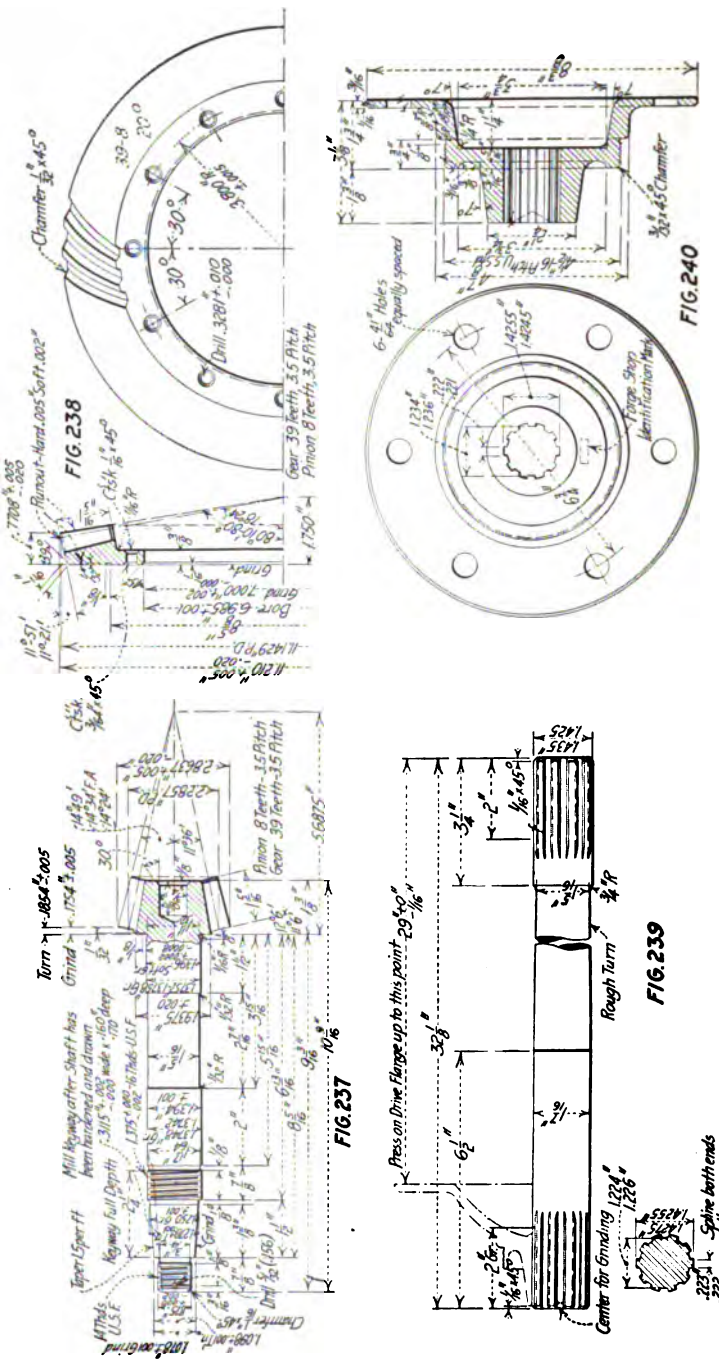


FIG. 238
Gear 39 Teeth, 3.5 Pitch
Pinion 8 Teeth, 3.5 Pitch
Drill .368" +.010 -.000
Chamfer 1/32" x 45°
FIG. 240
Chamfer 3/32" x 45° Chamfer
6-41" Holes equally spaced
Flange Shop Identification Mark
FIG. 237
Pinion 8 Teeth-3.5 Pitch
Gear 39 Teeth-3.5 Pitch
Mill Keyway after Shaft has been hardened and drawn
Keyway full Depth
Drill 5/16" (.312)
Chamfer 1/16" x 45°
FIG. 239
Press on Drive Flange up to this point 29 +0 -1/16"
Rough Turn
Center for Grinding
Spline both ends the same

FIG. 237.—Pinion. FIG. 238.—Ring gear. FIG. 239.—Drive shaft. FIG. 240.—Drive flange (Columbia Axle Co.).

is contracting and the internal expanding brake is toggle operated, as seen in the upper view of the assembly. The internal brake is operated by brake shaft *a* to which short lever *b* is attached. The external brake is actuated by brake shaft sleeve *m* and lever *n*, which operates the outer brake band lever *o* through link *g*.

The spring seats *s* are integral with the brake disk flange when using the Hotchkiss drive, and are pressed onto the housing and riveted in place as seen in the right side in the assembly. With a torque arm design a swivel type of spring seat is employed. With such a construction a boss is provided on the differential carrier to attach the torque arm thereto.

Lubrication.—The gears and pinion are operating in a bath of oil as previously mentioned, but the wheel bearings are packed in grease.

Figure 241 is a differential manufactured by the Warner Gear Co. as used on the Apperson car (which has an eight-cylinder engine 3 ¼- × 5-in., piston displacement about 330 cu. in.) The differential case hubs have hardened steel sleeves pressed in them for Hyatt roller bearings, at the points marked *s* in this figure as well as in Fig. 242, which is a detail of the differential case, flange side. This housing is made of malleable iron. There are 12-$^{21}/_{64}$-in. holes in the larger half to which the drive gear is riveted, and eight $^{25}/_{64}$-in. holes for assembling the two halves together by means of ⅜-in. studs, as seen in the assembly. In some designs, the two halves of the case are bolted together, while in others, sufficient metal thickness is provided to tap the holes in one half for cap screws.

The spiral bevel pinion and drive gear are seen in the assembly. The Warner Gear Co. state that for passenger car service, where extreme quietness is desired, they discourage the use of a pinion with less than 10 teeth, as they cannot be depended upon in regular quantity production to be sufficiently quiet when in use. Another suggestion made by this concern for the quiet running of gears is to make the bevel ring gear with a flat back whenever possible and with a very heavy section of metal under the teeth. They recommend that the amount of metal at the outer edge of the bevel teeth should not be less than the full depth of the tooth. By making the bevel gears heavy as suggested, they can be produced satisfactorily in quantity production to insure quiet and smooth running. Noisy gears may also be due to a light or weak construction of the differential carriers and portions

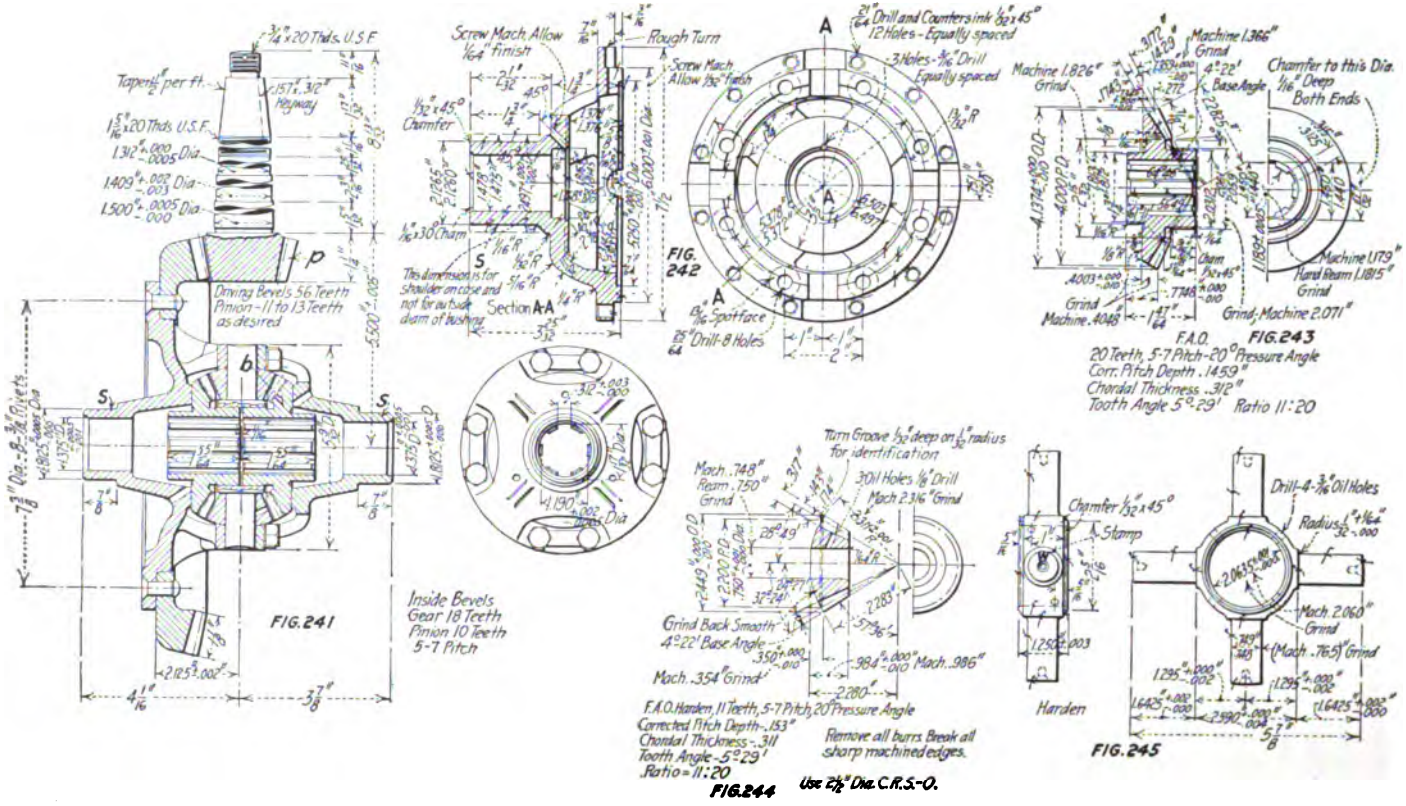


Fig. 241.—Warner Gear Co.'s differential. Fig. 242.—Differential case (flange side). Fig. 243.—Differential bevel side gear. Fig. 244.—Differential bevel pinion. Fig. 245.—Differential spider.

of the rear axle, which have to support the bearings; this is especially true where the spiral drive is used.

Figure 243 is one of the differential bevel side gears, (shown at *b* in the assembly), which is a drop forging. Note the accuracy to which the various dimensions are finished and ground. These bevel gears have 20 teeth, 5–7 pitch and a 20° pressure angle.

Figure 244 is a detail of the differential bevel pinion, of which four are used, which mesh with the two side gears. These pinions are made of S.A.E. 1020 steel, hardened; they have 11 teeth and are mounted on the differential spider (Fig. 245), which is a drop forging; after machining it is hardened and ground. It may be noted that a clearance of $\frac{1}{1.000}$ to $\frac{3}{1.000}$ in. is given these bevel pinions for a running fit over the arms of the differential spider. The oil holes drilled in the pinions and in the spider may be seen from the drawings.

Figure 246 is the rear axle (Salisbury) of the Paige-Detroit six-cylinder (3¼- × 5-in.), model 6-42 passenger car. The axle is of the three-quarter-floating type. The wheel bearings are double row new departure, which take the thrust in both directions; both, the lower as well as the upper race being clamped. At the inner ends of the shafts, supporting the differential, Timken taper roller bearings are provided; the spiral bevel pinion has a Hyatt roller bearing immediately in front of it where the radial load is greatest, at the front end of the pinion shaft a double row ball bearing is located which takes the radial load as well as the thrust in both directions.

Figure 247 is the housing assembly; the center of the axle housing is made of hot rolled steel $\frac{5}{32}$ in. in thickness; at the flanges *f* the metal is turned over for added strength and stiffness. The rear axle tube forming the ends of the rear axle housing, is made from steel tubing of 2½ in. outside diameter, thickness of wall ¼ in. and it is riveted to the axle housing by 14-$\frac{5}{16}$-in. rivets, as shown.

Figure 248 is a detail drawing of the differential carrier, a malleable casting, which is attached to the front of the housing by 12-⅜-in. cap screws with lock-washers underneath the heads. This carrier also supports the pinion shaft bearings (see Fig. 246).

The housing cover *T* at the rear of the axle is a stamping, made of hot rolled steel, 13 U.S.S.gage (.09375 in.). It is attached to the housing by 12,-⅜-in. cap screws, the same as the carrier; to render the housing oil tight, cork gaskets, $\frac{1}{16}$ in. thick, are

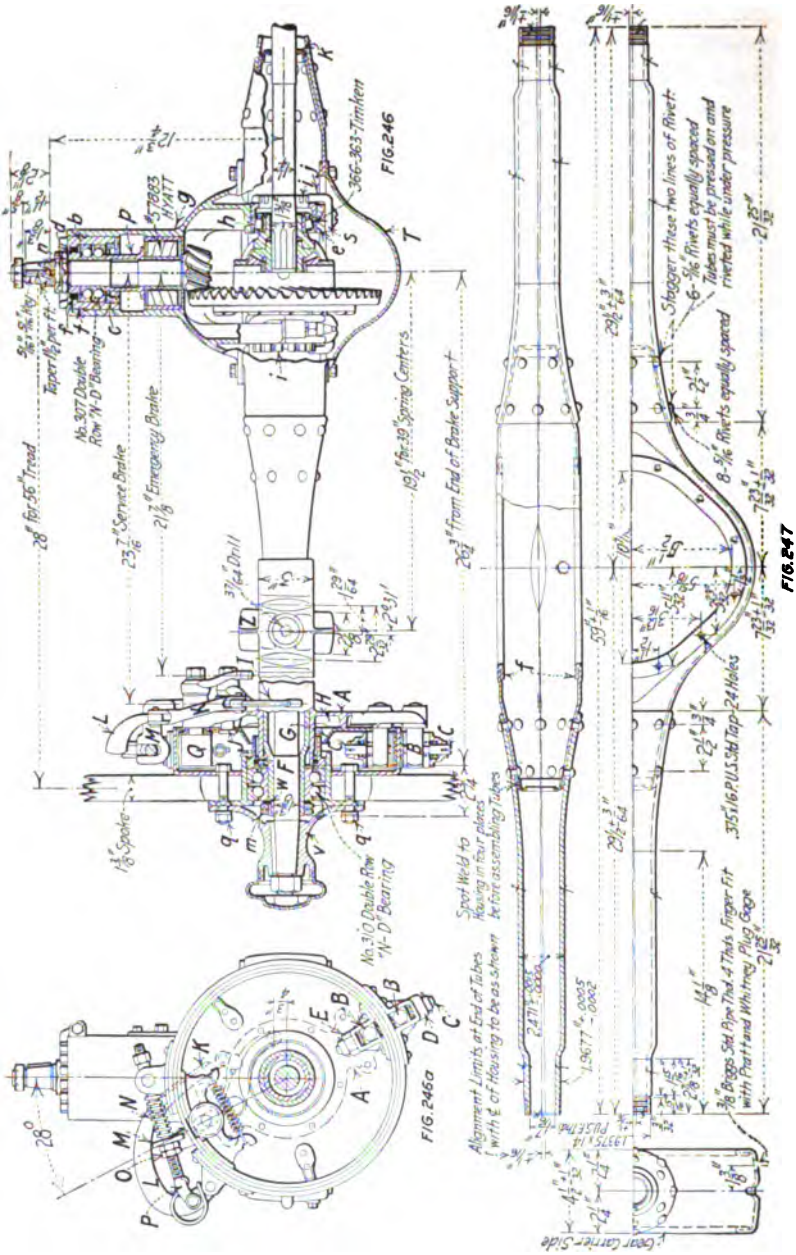


FIG.246
FIG.246a
FIG.247
No. 307 Double Row "N-D" Bearing
No. 310 Double Row "N-D" Bearing
Sport Weld to Housing in four places before assembling Tubes
Alignment Limits at End of Tubes with ℄ of Housing to be as shown
Stagger three lines of Rivets
6-5/16" Rivets equally spaced
Tubes must be pressed on and riveted while under pressure
8-5/16" Rivets equally spaced
24 Holes
Gear Carrier Side

FIG. 246.—Rear axle (Salisbury) of Paige-Detroit car. FIG. 246a.—Brake Mechanism. FIG. 247.—Housing assembly.

inserted between the joints of the housing and the differential carrier and housing cover. The gasket is shellacked to the housing (not to the cover) when assembled.

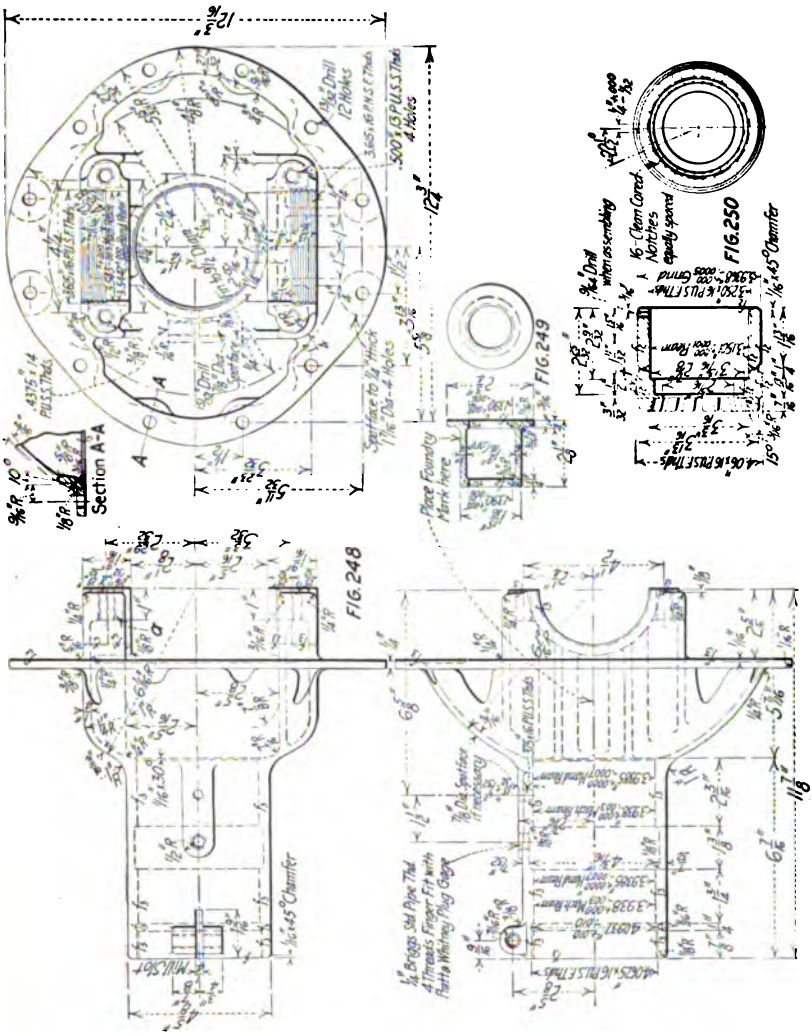


FIG. 248.—Differential carrier. FIG. 249.—Pinion bearing spacer. FIG. 250.—Bearing adjustment gage.

Figure 251 gives an assembly, with dimensions, of the differential and the spiral bevel final drive gears, and details of the spiral bevel pinion (Fig. 251*a*). The diameter of the pinion shaft is 1¼ in. at the large end of the taper. The pinion has eight

teeth, the drive gear 38, hence the ratio in the rear axle is $\frac{38}{8}$ = 4.75 to 1. The pinion, having a right hand spiral, mates with

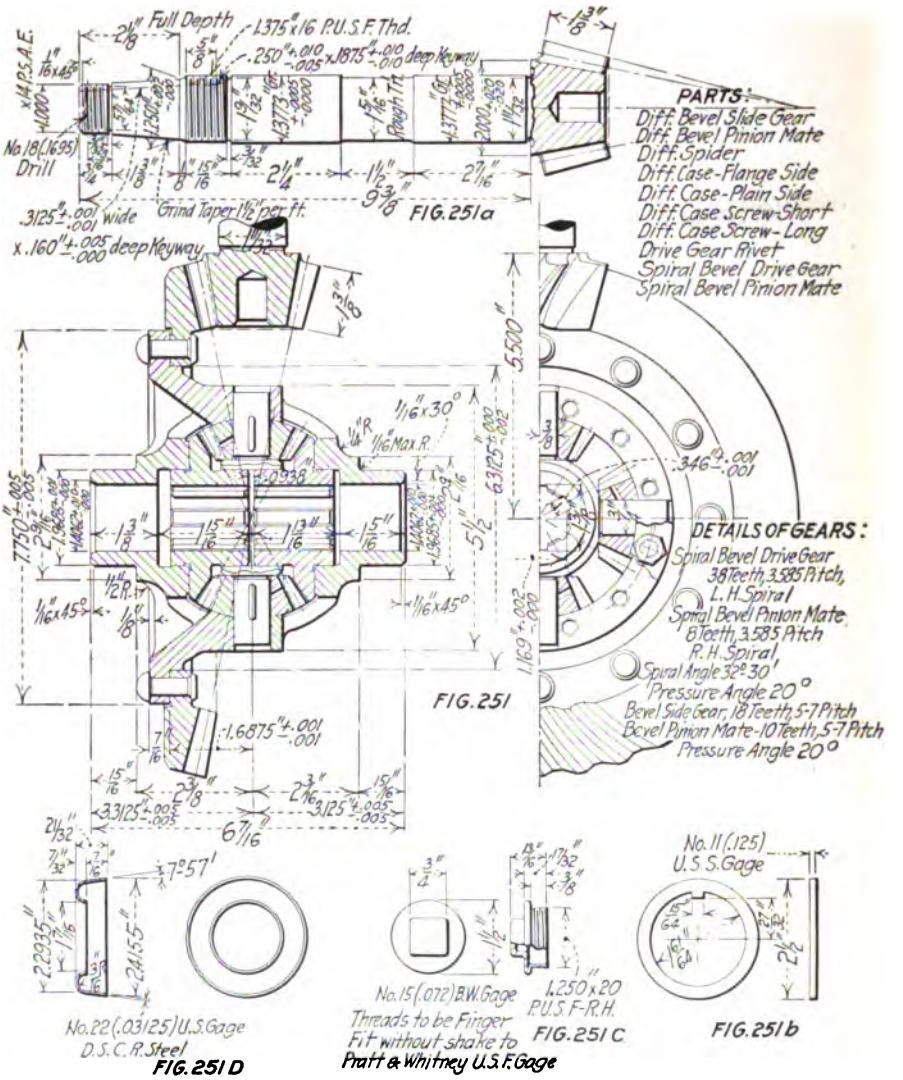


FIG. 251.—Differential assembly (Page-Detroit, Salisbury type). FIG. 251a.—Spiral bevel. FIG. 251b.—Lock washer. FIG. 251c.—Grease plug. FIG. 251d.—Grease retainer.

the left hand spiral of the drive gear. Their pitch is 3.858, the spiral angle 32° 30', the pressure angle 20°, the face width of

both 1⅜ in. A Hyatt roller bearing is located at the pinion end of the pinion shaft (see Fig. 246) abutting against the shoulder of the pinion, at the rear, and against the pinion bearing spacer *p* at the front. Figure 249 is a detail of this spacer made of cast iron, having a wall thickness of 3/16 in. A double row, new departure 307 bearing, is located in front of the spacer. This ball bearing and the pinion shaft complete are assembled in the bearing adjustment cage (front bearing adjuster) *b*, see detail (Fig. 250) in which the upper race is clamped by the bearing retainer *c*. A detail of this bearing retainer is shown in Fig. 254. Both the bearing retainer and the adjuster are malleable castings. By means of this adjuster and the nuts *n*, the pinion and the bearings are held in place in the gear carrier *g* by threaded portion *t*; the adjuster being locked in place in the carrier. The lower race of the ball bearing is clamped by the nuts *n*, through the intermediary of spacer *d*. A felt washer *f* prevents oil leaks and forms an efficient enclosure for the bearings against dirt and grit.

Figure 252 is a detail of the differential bearing cap *e*, by means of which the outer race of the differential roller bearing *h* is clamped to the gear carrier; ½-in. studs being used to hold the cap in place, the carrier being threaded for a length of 1 in. (marked *a* in Fig. 248) for these studs. The taper roller bearings are adjusted by means of differential bearing adjusters (adjusting collars) *i*, which are locked by the adjuster locks *j*, (see assembly Fig. 246) a detail of which is shown in Fig. 253. One end of this locking finger is placed between the notches of the adjusting collar, the other, inserted into the slot of the cap at *b*, (Fig. 252), and then locked by means of pin *s*. A detail drawing of the differential bearing adjuster is given in Fig. 253*a*.

The axle shaft (drive shaft or live rear axle) is made of chrome-nickel steel 1⅜ in. in diameter in the center. It has six splines to correspond with those of the differential side gear, a taper fitting and key for attachment to the driving wheel flange and it is drawn unto its seat by a 1-in. nut. By removing the nuts *q*, the shaft and drive flange *v* may be removed, thus exposing the nuts *m* which clamp the lower race of the double row new departure 310 wheel bearing. By removing these nuts the wheel can be taken off. Between the lower race and the first nut *m* there is a lock washer (Fig. 251*b*) the prong of which fits into the milled slot at *w* (see assembly). There is another lock

washer between the two nuts *m*, a detail of the latter is shown on Fig. 255.

The spring seat, a malleable casting, is attached to the axle tube, (*Z* in the assembly).

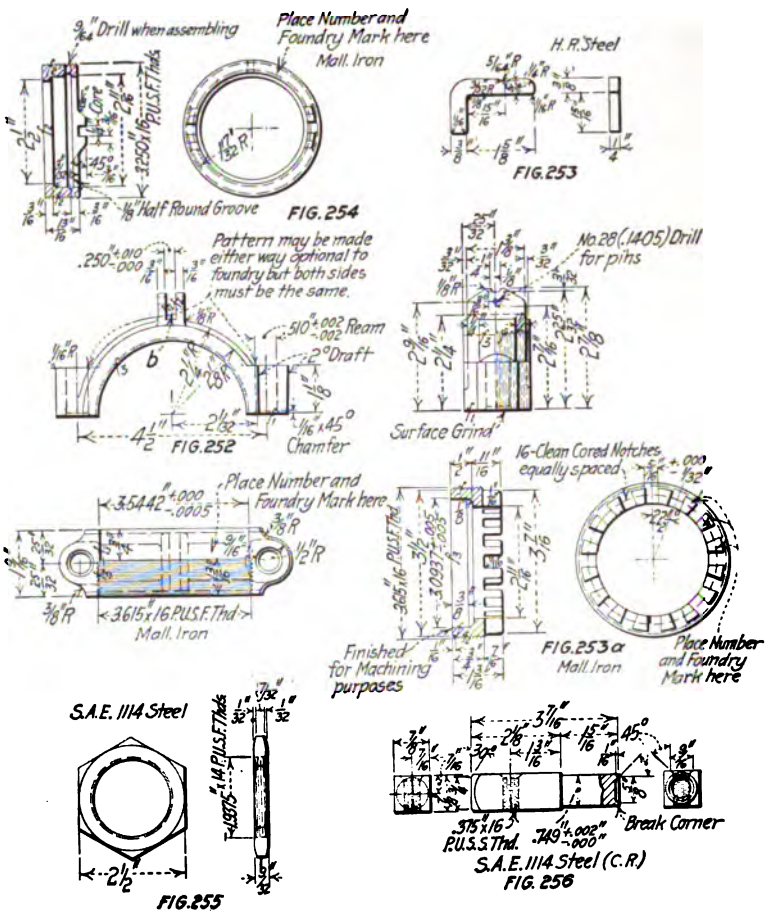


FIG. 252.—Bearing cap. FIG. 253.—Adjuster lock. FIG. 253a.—Bearing adjuster. FIG. 254.—Bearing retainer. FIG. 255.—Nut. FIG. 256.—Brake band support stud.

Figure 251*d* is the grease retainer for the housing noted, marked *k* at the right hand end in the assembly drawing. Figure 251*c* is the grease plug for the housing cover; it is a steel stamping.

Brake Mechanism.—Figure 246*a* shows the brake assembly of the Paige-Detroit rear axle (Salisbury type) disclosing the

internal expanding and the external contracting brake bands, which are of the wrap-up type.

Figure 257 is the brake support (a malleable casting) seen at *A* in Fig. 246, and also dotted in Fig. 246*a*. This brake sup-

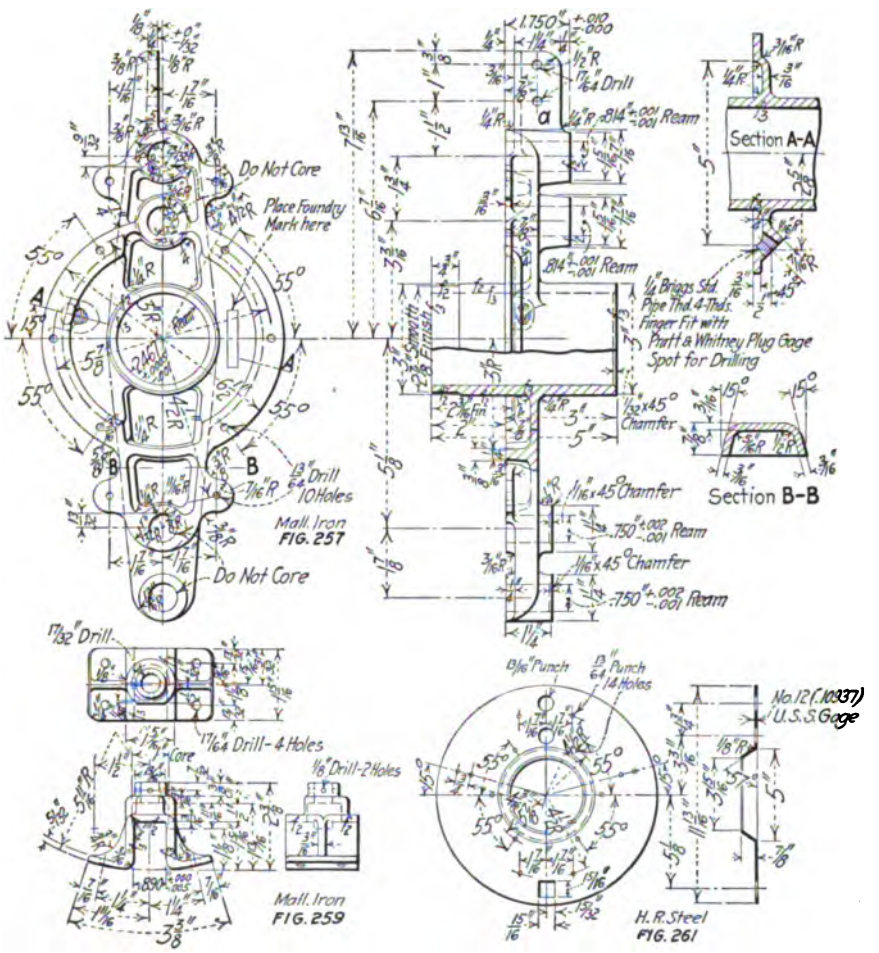


FIG. 257.—Brake support. FIG. 259.—Internal brake band support. FIG. 261.—Brake drum disk.

port is attached to the axle tube (driven on it and welded to it) as seen from the assembly. The diameter of its hub is reamed as indicated on the detail to .004 in. less than the outside diameter of the tube to which it is attached. One end of its hub clamps

the wheel ball bearing. This brake support is made very strong and yet light in weight, by making it U-section wherever possible; as it must carry the entire torque from the wheels when applying the brakes, it has to be made very rigid.

Figure 256 is a detail of the brake band support studs, of which two are required for, and are riveted to, each brake support, as indicated at *B*. Figure 258 is the brake band adjusting screw, which is threaded into, and passes through, the stud, and by means of it the clearance between the brake drum and the brake bands is adjusted to the required amount, about $\frac{1}{32}$ in. Figure

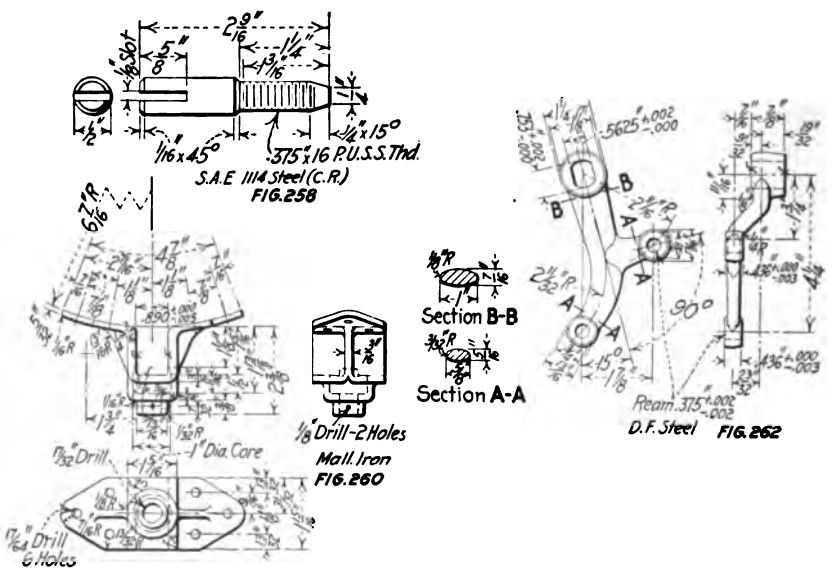


FIG. 258.—Adjusting screw. FIG. 260.—External brake band support. FIG. 262.—External brake lever.

259 is the support for the external brake band, indicated at *D*, and Fig. 260, the internal brake band support, marked *E* in the assembly. They are riveted to the external and internal brake bands respectively. The square end of the brake support stud fits into them, thereby holding the brake bands in place. Both these brake band supports are malleable castings, and are riveted to the brake bands by $\frac{1}{4}$-in. rivets. Figure 261 is the brake disk, which covers the inner side of the brake drum (indicated at *G*) thereby enclosing the internal brake mechanism, and it also acts as an oil catcher, in conjunction with the brake sup-

port. Any oil, which should flow through the rear axle housing and into the wheel bearings, through these bearings and past the felt washer *F*, would flow into the cavity formed at *H*, and is thereby prevented from reaching the internal brake band. This brake disk is made of No. 12 U.S.S. gage (.10937 in.) hot rolled steel. The square hole at the bottom, fits the square end of the internal brake band support stud, while the upper two holes accommodate the brake lever shafts. Figure 262 is the brake lever for the external brake; its shaft *J* oscillates in the brake support and fits into the oblong hole of the lever to which it is rigidly attached. Figure 263 is the brake connecting link, a steel stamping, (*K* in the assemblies) which connects the brake lever with the outer brake contracting lever *L*, a detail of which is given in Fig. 264. Note how the section area decreases from the fulcrum to the end of the lever, as the leverage, or the moment of the force, and hence the stress in the metal, is greatest near the fulcrum. Figure 265 is the brake retainer, marked *M*, (Figs. 246 and 246*a*), by means of which the two halves of the external brake band are held away from the brake drum; the spring *N* holds one-half clear, and the nuts *O* the other half, by means of the threaded fork *P*. The brake retainer *M* is riveted to the brake support at *a* in the right hand view of Fig. 257.

Figure 266 is a detail of the brake lever for the internal brake, which is rendered adjustable by the 40 serrated teeth. It is rigidly attached to the shaft of cam Q, which operates the internal brake by forcing its two ends apart, to which the brake band ends, (Fig. 267), are riveted. When the lever is actuated, the cam Q, is applied against the flat surfaces *b*, thereby forcing the brake band open. Figure 268 is a detail of the cam and internal brake shaft, they being made integral. The external brakeshaft *J* is shown in Fig. 269; it is made of S.A.E. 1114 steel.

The external brake band (12$\frac{5}{16}$ in. in diameter), and the internal brake band, (Fig. 270), are made of .093-in stock. The asbestos lining riveted to these bands is 2 in. wide and $\frac{5}{32}$ in. thick, and the rivets used are $\frac{5}{32}$ in.; the holes in the bands being $\frac{1}{64}$ in. larger.

Figures 271 and 272 show the construction of the rear axle of the Packard Twin-Six passenger car (which has a twelve-cylinder engine, 3- × 5-in.) having a ratio of 4.36 to 1 and a wheelbase of 136 in. The mounting of the pinion and the differential gears can be seen from the drawings. For the pinion, a double row bearing

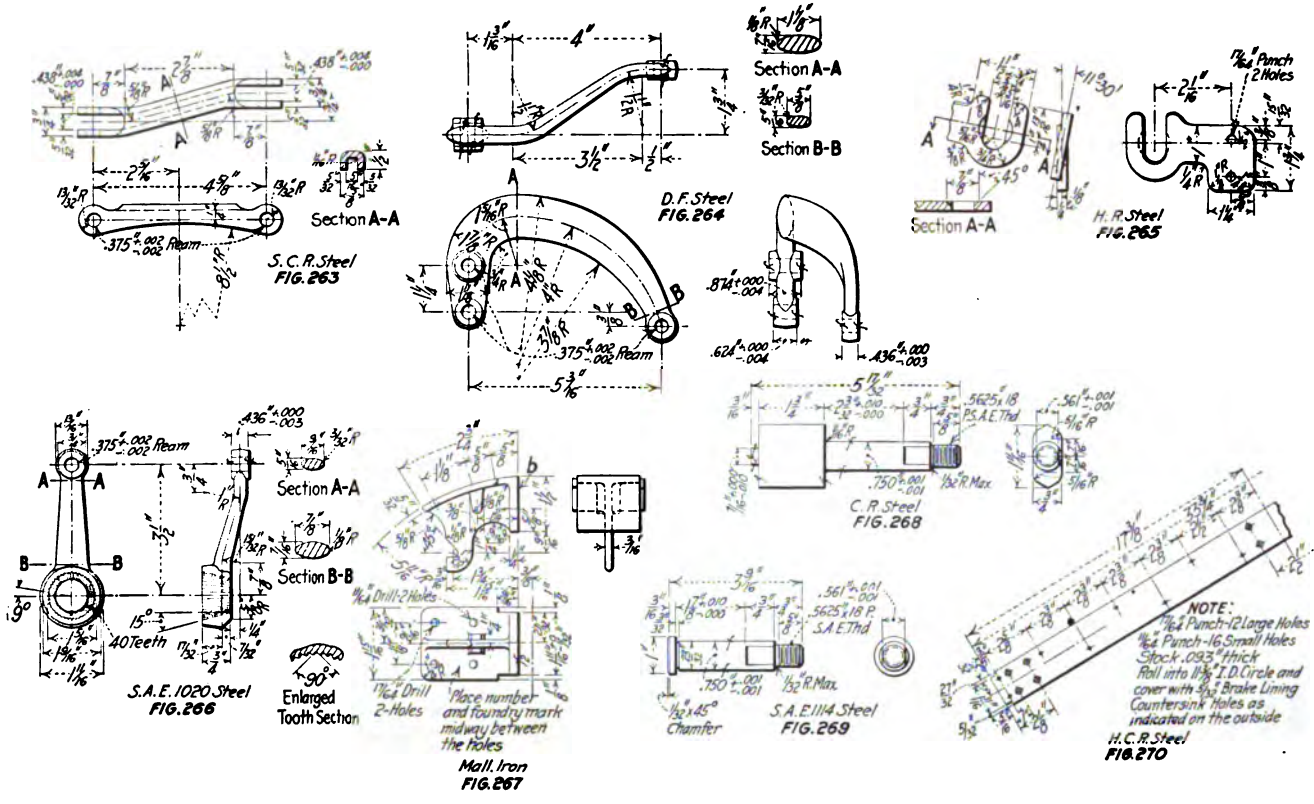


FIG. 263.—Brake connecting link. FIG. 264.—Contracting lever. FIG. 265.—Brake retainer. FIG. 266.—Internal brake lever. FIG. 267.—Brake band ends. FIG. 268.—Brake cam. FIG. 269.—External brake shaft. FIG. 270.—Internal brake band.

is used in the front, to take the radial as well as the thrust load in both directions; the rear pinion bearing is located between the pinion and the differential case. The differential housing is supported on single row ball bearings with ball thrust-bearings at the back for the thrust loads. The wheels, fixedly attached to the semi-floating axle, are supported on a single row ball bearings having their upper and lower races clamped to take the thrust in both directions.

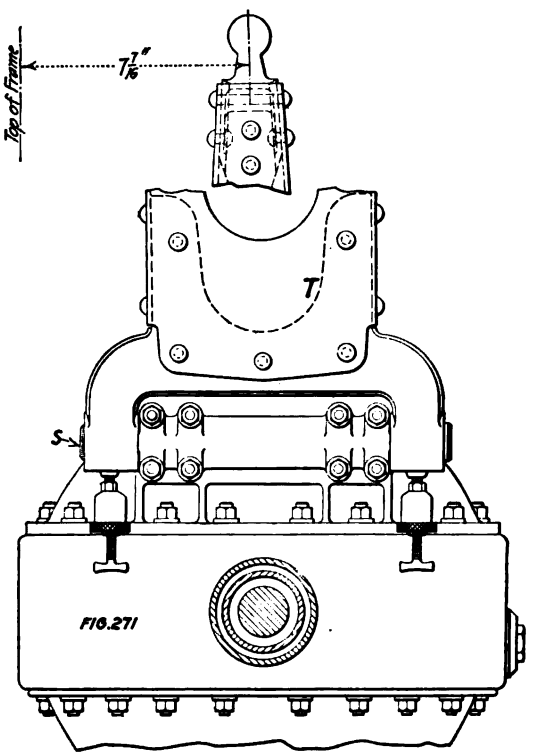


FIG. 271.—Packard (Twin-Six) torque arm.

The torque arm *T* is of channel section and is hingedly attached to the axle housing by shaft *S*. The internal expanding brake consists of cam operated shoes. The Packard Single-Six car is of later design than the Twin-Six, and uses the Timken type of semi-floating rear axle.

Figure 273 shows the construction of the three-quarter floating, Haynes car, model 75 rear axle; Fig. 274 is a plan view of the com-

plete chassis, which has a wheelbase of 132 in. and a six-cylinder engine of $2\frac{1}{2}$- × $5\frac{3}{16}$-in., that is, a piston displacement of 299 cu. in. In the rear axle ball bearings are used throughout. The pinion is supported between two sets of double row ball bearings; the wheel on two single row bearings. A ball thrust-bearing is located on one side of the differential to take the thrust from the spiral bevel drive. At *p* is the location of the pin by

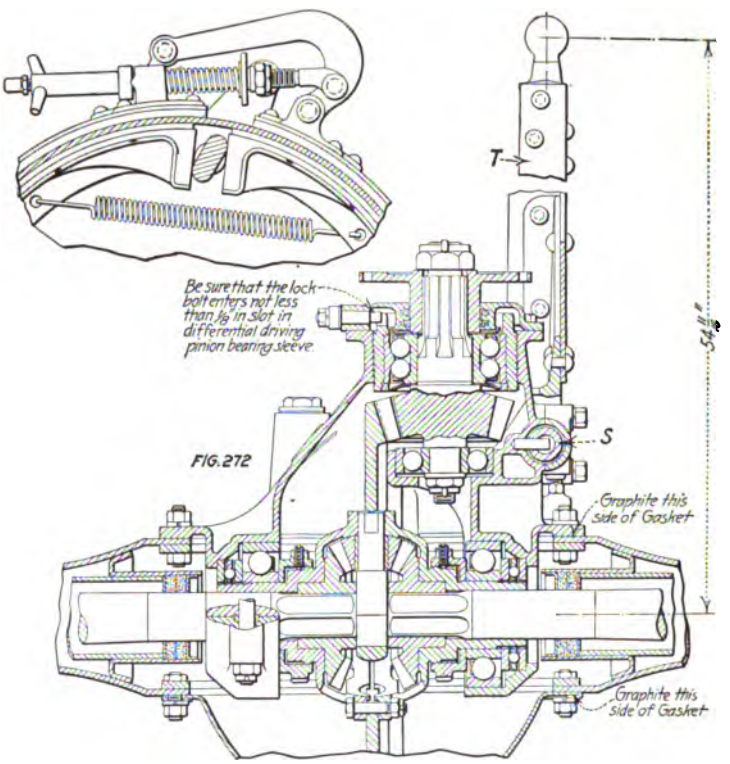

T
Be sure that the lock-bolt enters not less than 1/8 in slot in differential driving pinion bearing sleeve.
FIG. 272
S
Graphite this side of Gasket
Graphite this side of Gasket

FIG. 272.—Packard twin-six rear axle.

which the torque arm is attached to the rear axle. The torque arm can be seen at *t* in Fig. 274. The gear reduction in the rear axle is 4.6 to 1.

Figure 275 is an assembly of the Timken rear axle for passenger cars weighing approximately 3,800 lbs. The housing is of reinforced pressed steel. The pinion is supported between Timken tapered roller bearings; the differential and the wheels

are also supported on this type of bearing. The axle is of the semi-floating or fixed hub type. The brakes are internal expanding and external contracting, (see Fig. 276). The brake drum has an outside diameter of 15½ in. with an overall width of 2¾ in., providing brake bands 2½ in. wide. Note at *A*, in the center of

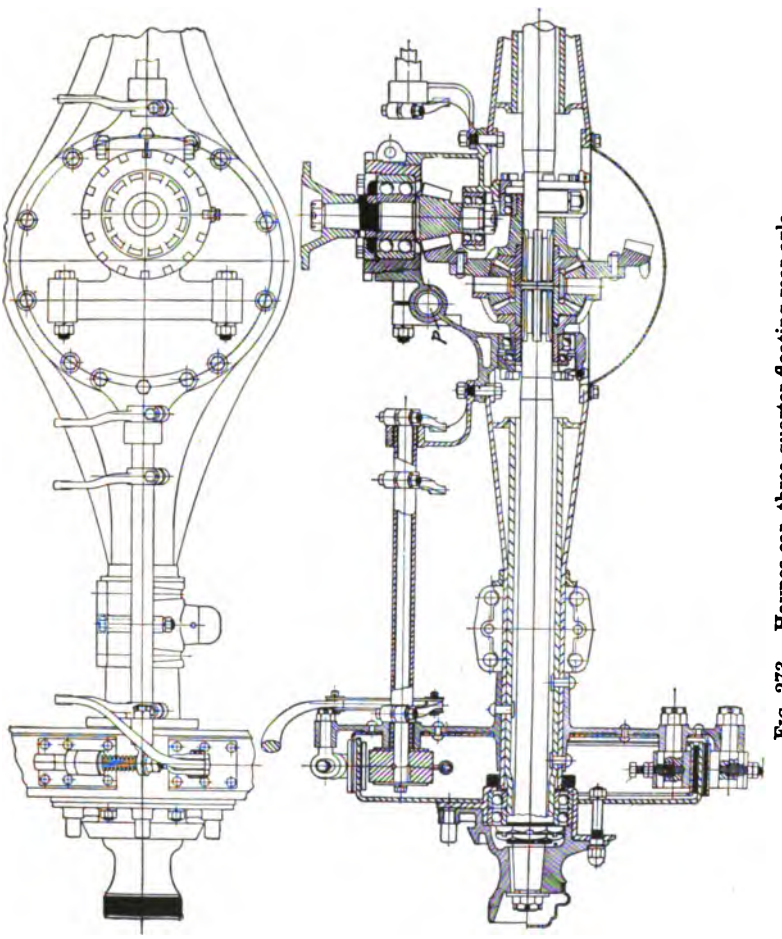


FIG. 273.—Haynes car, three-quarter floating rear axle.

the axle, the hardened plugs inserted in the drive shaft where they meet. The diameter of the plugs being 9/16 in. The object of these is to transfer the outside thrust from one wheel to the bearing of the next wheel. Figure 277 is a detail of the drive shaft, showing the hole for the said plug on the right hand side.

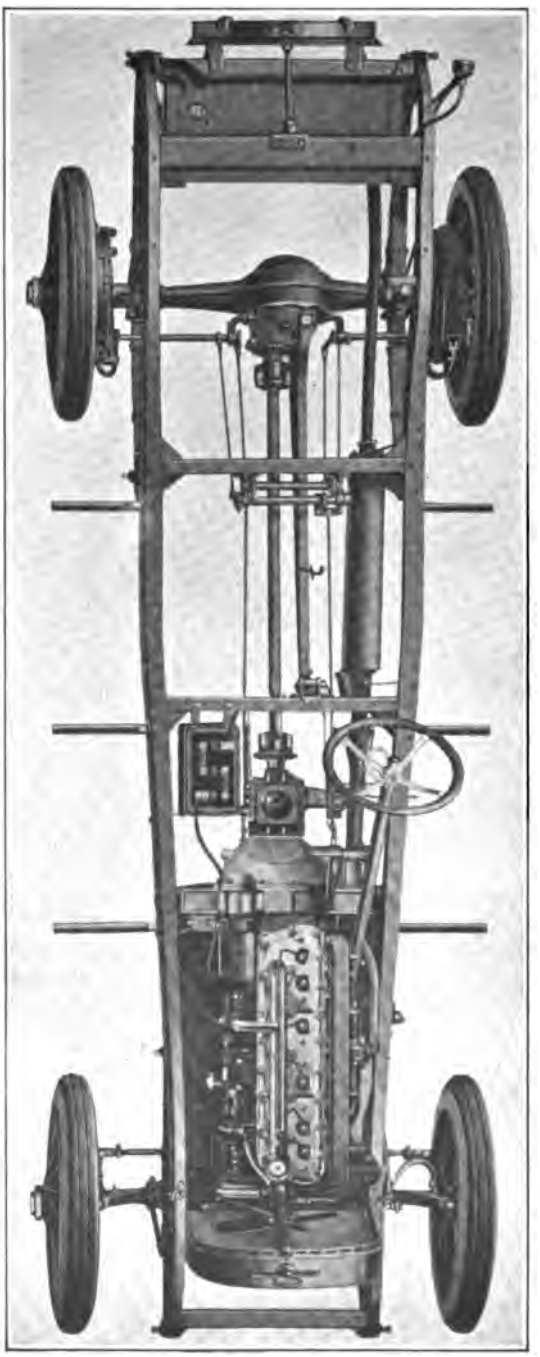


FIG. 274.—Haynes passenger car chassis.

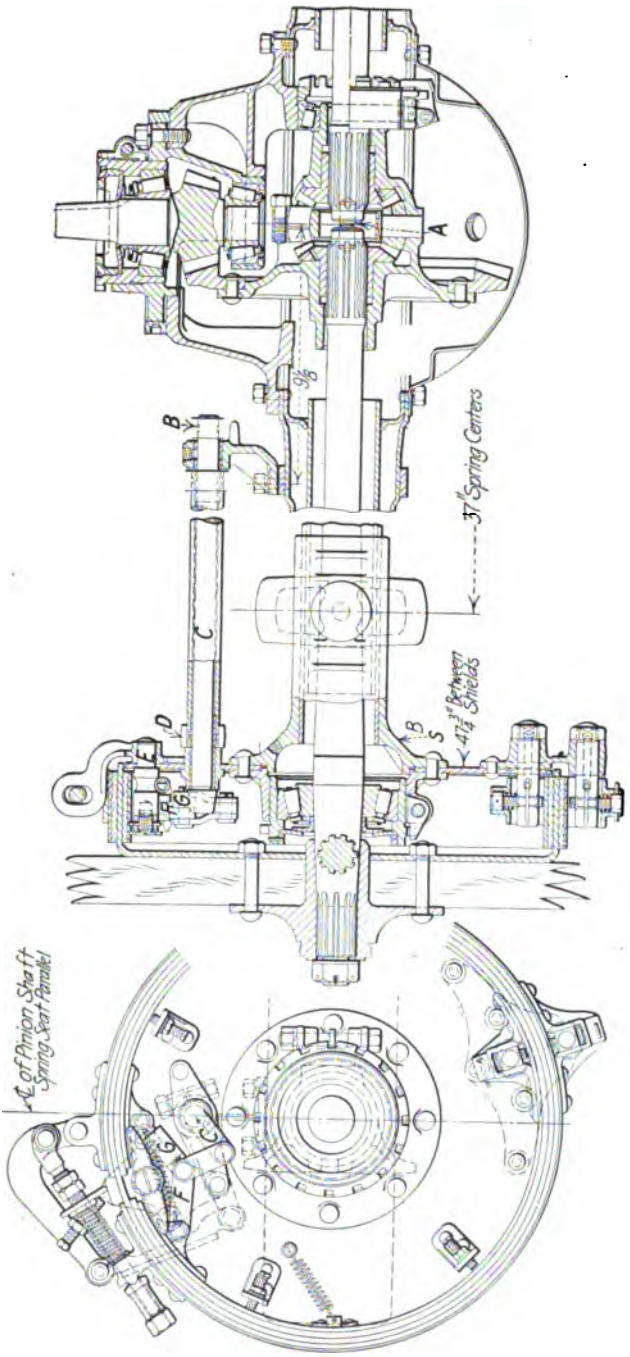


FIG. 275.—Timken rear axle (semi-floating) for passenger cars. FIG. 276.—Brake mechanism.

The shaft is made of S.A.E. 3140, chrome-nickel steel, heat treated.

Figure 278 is a detail of the gear and pinion. The gears have helical teeth, the pinion being cut with a left hand spiral and the bevel gear with right hand spiral. The pitch is 4.25 in. and the reduction ratio $4\frac{5}{11}$ in. The gears are made of special steel, (chrome,-vanadium,-nickel) carbonized $1\frac{1}{32}$ in. deep, and heat treated.

Figure 279 is a detail of the rear wheel hub. All the dimensions and tolerances are given. The flange is provided with square holes for attachment to the wheel.

The eight-cylinder Cadillac car as well as the Mercer have their pinions supported in a similar manner, as the Timken shown in Fig. 275, *i.e.*, between two roller bearings, but the Cadillac has a full-floating axle supported on two rows of roller bearings, while the Mercer full floating is supported by two single row ball bearings. In the Cadillac rear axle the torque is not taken by the springs but by a torque arm fastened to a central cross-member of the frame, like that shown in Fig. 274, *t* being the torque arm.

Figure 280 shows the rear axle used on the Dodge car, which has a four-cylinder engine of $3\frac{7}{8}$-in. bore by $4\frac{1}{2}$-in. stroke and a wheelbase of 114 in. In this axle a torque tube is used concentric with the propeller shaft, and extends to the universal joint behind the transmission. This axle is of the three-quarter floating type; the type of three-quarter floating, where two bearings are employed for supporting the wheel, are sometimes called seven-eighth floating.

The rear axle of the Essex car is of similar design to that of the Timken rear axle described in Fig. 275, except that the pinion is located behind the two bearings as seen from Fig. 281; the dimensions of the different parts are smaller, but in other respects the design is practically identical. The pinion and gear are made of chrome-vanadium-nickel steel, carbonized. This car has an engine of $3\frac{3}{8}$-in. bore by 5-in. stroke. The housing is of reinforced pressed steel, the driving gear and the differential being mounted on a separate carrier which is bolted to the axle housing. The brake bands in the Essex are 14 in. in diameter and $1\frac{3}{4}$ in. wide.

Figure 282 is a detail of the rear axle brake spider. Its location is indicated at *S* in Fig. 275, but this and the following detailed drawings are for a car of the capacity of the Essex,

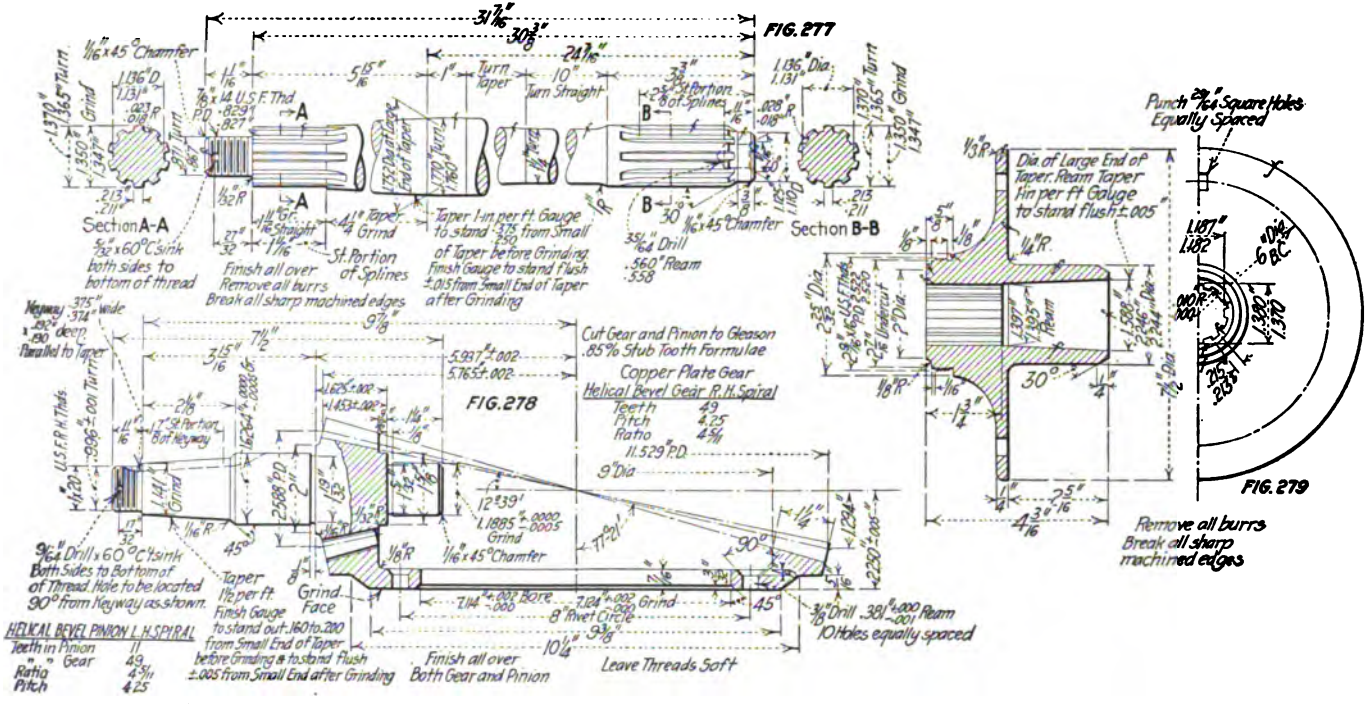


FIG. 277
FIG. 278
FIG. 279
Section A-A
Section B-B
Finish all over
Remove all burrs
Break all sharp machined edges
Taper 1 in per ft Gauge to stand .375 from Small End of Taper before Grinding
Finish Gauge to stand flush ±.015 from Small End of Taper after Grinding
Cut Gear and Pinion to Gleason .85% Stub Tooth Formulae
Copper Plate Gear
Helical Bevel Gear R.H. Spiral
Teeth 49
Pitch 4.25
Ratio 4 11/49
11.529" P.D.
9" Dia
Finish all over Both Gear and Pinion
Leave Threads Soft
Punch 29 13/64 Square Holes Equally Spaced
Dia of Large End of Taper. Ream Taper 1 in per ft Gauge to stand flush ±.005"
Remove all burrs
Break all sharp machined edges
HELICAL BEVEL PINION L.H. SPIRAL
Teeth in Pinion 11
" " Gear 49
Ratio 4 11/49
Pitch 4.25

FIG. 277.—Drive shaft. FIG. 278.—Gear and pinion. FIG. 279.—Rear wheel hub (Timken rear axle).

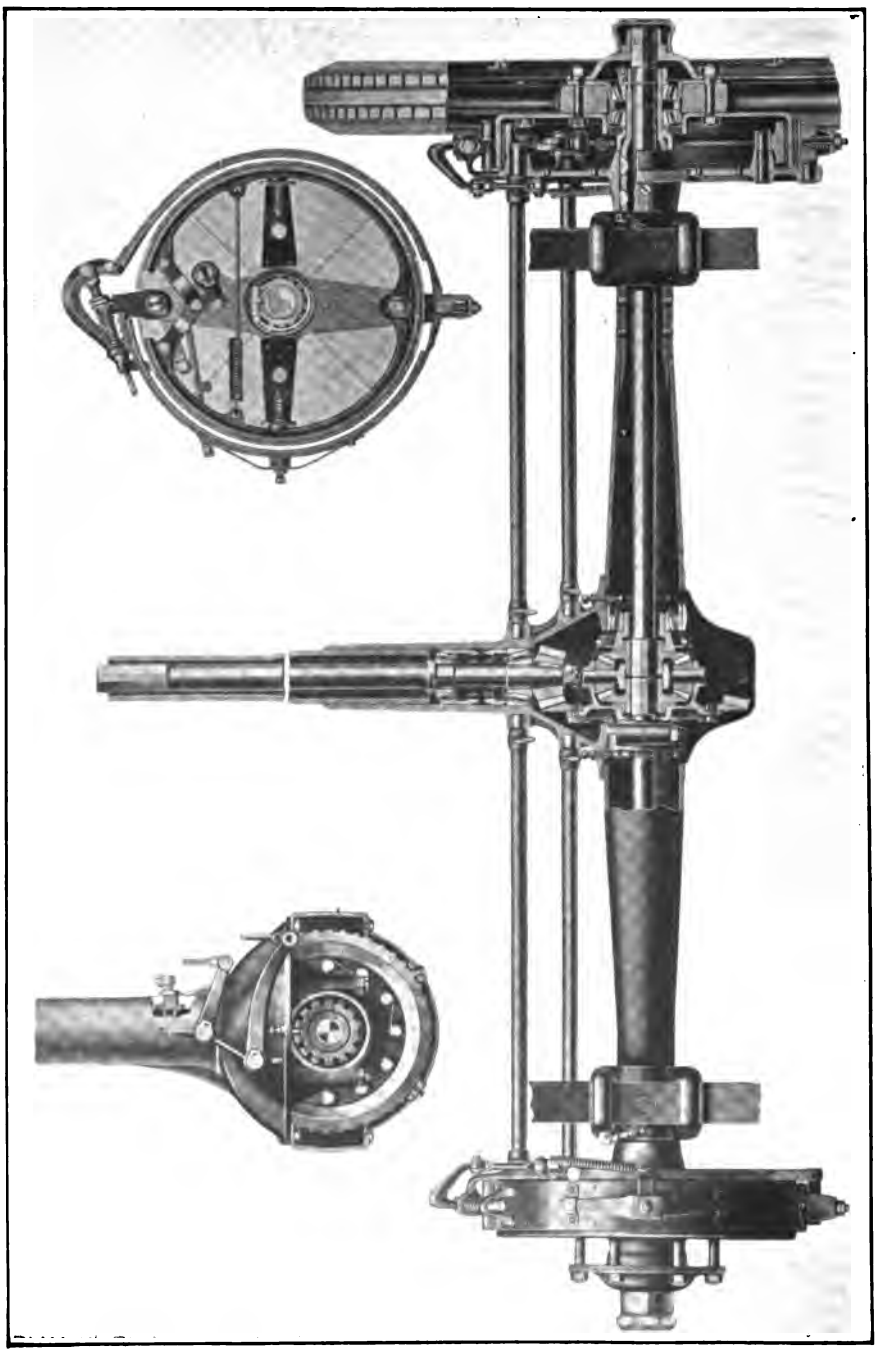


FIG. 280.—Dodge car rear axle.

having a weight, without passengers, of about 2500 lb. This brake spider is a malleable casting. The rear wheel hub is very similar in dimensions to that shown in Fig. 279, except that it is not splined but has a taper fitting with a key ¼ in. wide. The total length of the hub is $3\frac{7}{16}$ in., the taper at the small end in the hub 1.0944 in., and in the large end 1.3809 in.

Figure 283 is the internal brake cam shaft marked *C* in Figs. 275 and 276. It is made either of drop forged steel or cold drawn steel, S.A.E. 1035, the lever end being case hardened as

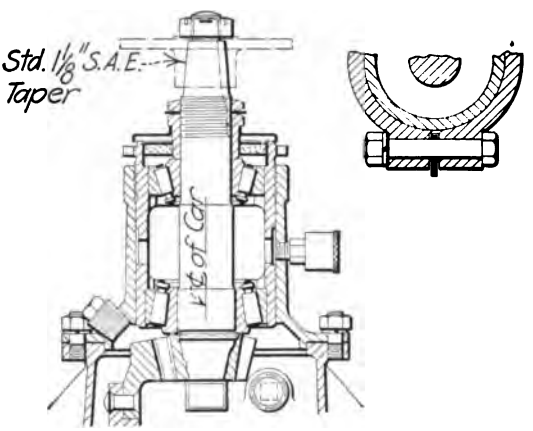


FIG. 281.—Pinion mounting of Essex car rear axle.

indicated in the drawing, *i.e.*, it is heated to 1,450°F. in cyanide and then quenched in water.

Figure 284 is the lever of the external brake tube indicated at *D* in the assembly drawing. It is provided with serrations for the purpose of adjustment. Figure 285 is the internal brake guide pin made of bar screw steel S.A.E. 1114. The location of this part is indicated at *E*. Figures 286 and 287 are the internal brake expanding links located at *F* and *G* in the assembly drawings. The former is made of hot rolled steel S.A.E. 1020 while the latter is made of bar screw steel S.A.E. 1114.

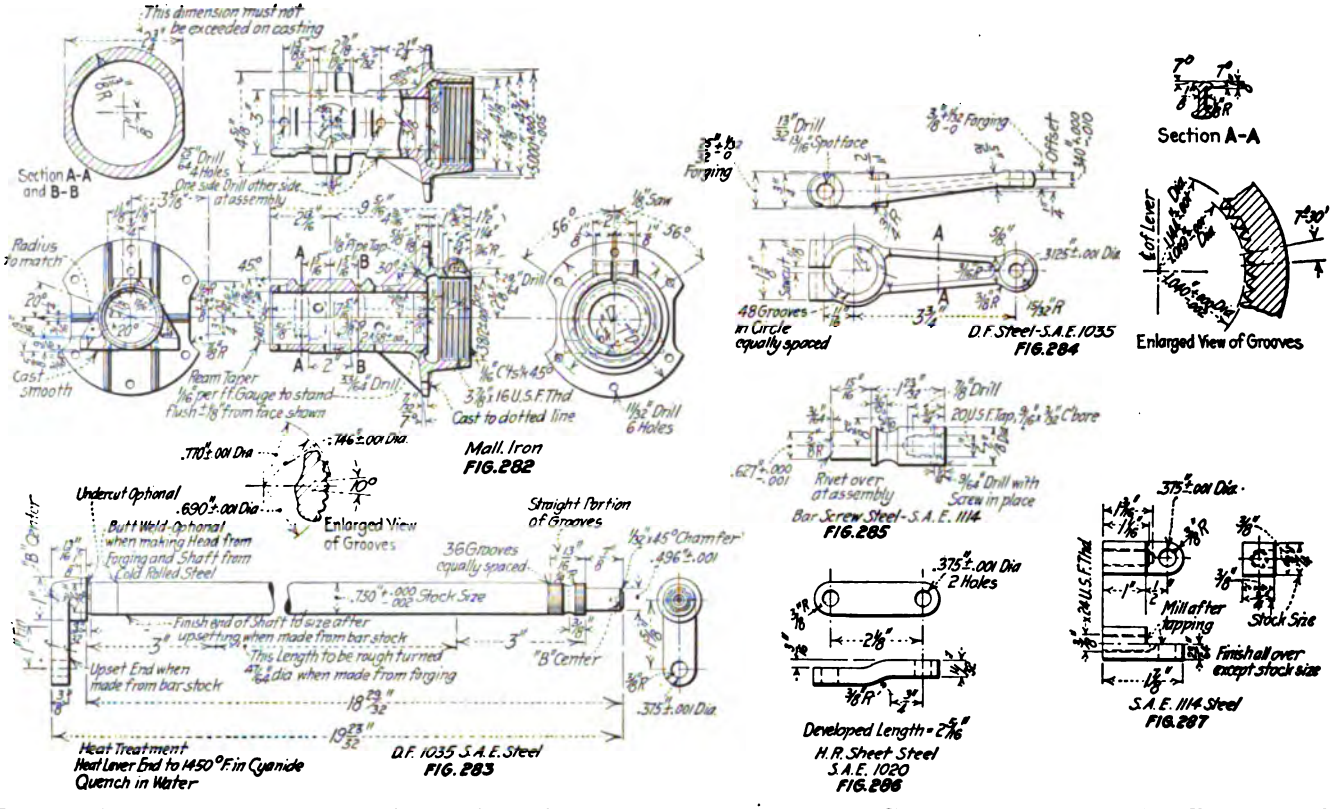

This dimension must not be exceeded on casting
Section A-A and B-B
Radius to match
Cast smooth
Ream Taper 1/16" per ft Gauge to stand flush ±1/8" from face shown
Cast to dotted line
Mall. Iron
FIG. 282
Undercut Optional
Butt Weld-Optional when making Head from Forging and Shaft from Cold Rolled Steel
Enlarged View of Grooves
Straight Portion of Grooves
36 Grooves equally spaced
Finish end of Shaft to size after upsetting when made from bar stock
This Length to be rough turned when made from forging
Upset End when made from bar stock
Heat Treatment
Heat Lever End to 1450°F. in Cyanide
Quench in Water
D.F. 1035 S.A.E. Steel
FIG. 283
Forging
48 Grooves in Circle equally spaced
D.F. Steel-S.A.E. 1035
FIG. 284
Section A-A
Enlarged View of Grooves
Rivet over at assembly
Screw in place
Bar Screw Steel-S.A.E. 1114
FIG. 285
2 Holes
Developed Length = 5 7/16"
H.R. Sheet Steel
S.A.E. 1020
FIG. 286
Mill after tapping
Stock Size
Finish all over except stock size
S.A.E. 1114 Steel
FIG. 287

Fig. 282.—Brake spider. Fig. 283.—Brake cam shaft. Fig. 284.—External brake lever. Fig. 285.—Brake guide pin. Figs. 286 and 287.—Expanding links (Essex rear axle).

CHAPTER VIII

REAR AXLES FOR TRUCKS

The rear axle is one of the most important parts; in trucks, it carries from 50 per cent to 90 per cent of the total load, transmits the power, carries the torque reaction on braking or on locking of the wheels and it is subjected to road shocks more severe than pneumatic tire shod passenger cars. When running with no load the springs of a truck deflect very little, hence the vibrations and jars are enormous.

Final Drives for Trucks (see page 192 for the different drives).—Each of the various types of final drives have their adherents, claiming advantages for certain types over those of others. The author will endeavor to cite the advantages and disadvantages of each type as put forward in the literature of the various makers, and examine the merit of the arguments. In passenger cars the bevel gear drive is employed almost universally, most companies using the spiral bevel type, on account of its greater silence in running, and because it permits the use of a smaller pinion. Since bevel gears are not as a rule practicable, for reductions in excess of about 6 to 1, their use on trucks except on the smaller models, is ordinarily out of the question, for here the final reductions vary from about 6½ to about 14, depending on the capacity of the motor and the truck speed.

Chain Drive—The chain drive on trucks is used on about 3.9 per cent of the models manufactured. The chief objection to the use of chains is their exposure to dirt and grit which in time causes excessive wear and noise. A number of attempts have been made to enclose the chains by covers of some kind, but they gave more or less trouble and were not entirely successful, for provisions must be made in the cover for chain adjustments and for relative lateral as well as vertical movement of the sprocket centers, on account of sidesway and spring flexure. Hence, to-day, most of the chain drives run with the chains exposed, and by the employment of specially hardened and constructed small sprock-

ets, chain rollers, pins, bushings and links, the wear and the noise are minimized.[1]

For the reasons mentioned, many joints and bearings have been found necessary in connection with chain covers, and practice has shown that, in service, the chain case will become frequently more noisy than the exposed chain. When considering the added cost of maintenance of the parts forming the chain-case and the time lost in the repair shops due to the inaccessibility of contiguous parts, it has been found less expensive in the long run, to leave the chain-case off, even though the chain life is thereby somewhat shortened. It is claimed that efficiency losses due to running roller chains exposed to dust and dirt are very small even after the chain has become worn.

Excessive noise and wear in the chain drive may be caused by the following:

(*a*) Too few teeth in driving sprockets.
(*b*) Improperly cut sprockets.
(*c*) Excessive misalignment.
(*d*) Lack of lubrication.
(*e*) Improper adjustment.
(*f*) Worn out chains and sprockets.

The first three are questions of design and workmanship, and these are not usually the cause of noise when such exist in the modern truck. Lack of lubrication is by far the most widespread reason for noisy chains. It is claimed that applying lubrication with a brush each day takes only a few minutes, and results in greatly decreased noise and wear. If chains are properly lubricated every day, it is claimed they will last from 40 to 50,000 miles, but users find the average life of chains about 20,000 miles. The following are the advantages claimed for the chain drive over the various other forms of final drive:—

1. Simplest in construction.
2. Will stand as much or more abuse than any other type.
3. Average total efficiency, including the losses in the jackshaft, is as high or higher than that of any other form of drive.
4. Increased losses due to worn chain and sprockets are very slight.
5. Easiest form of final drive to repair, as repairs can usually be made on the road with spare links.

[1] H. D. CHURCH. *S.A.E. Transactions*, 1914, Part I, and 1916, II,

6. Bevel and differential gears are carried on the sprung portion of the vehicle and are thus immune from road shocks.

7. A substantial load-carrying member can be employed, which in the event of an accident will not damage the drive-shaft and the gears.

8. The differential gears and the drive-shaft can be made lighter than in most of the other types as the maximum torque is produced only at the rear wheels.

9. The universal-joints are arranged to run in line between the transmission and the engine, since both are attached rigidly to the chassis.

10. Offers the lowest unsprung weight, thus increasing the riding comfort.

The International Motor Co., manufacturers of the Mack truck, use the chain drive exclusively for their heavier models but give the purchaser the option of a chain drive or a double reduction drive in the smaller sizes. While chains will require more frequent repairing and renewing than the gears of the worm or bevel drives, the chain drive is the easiest to repair. It also offers a greater road clearance than the live axle type of drive and a much greater range of gear ratio changes than any other type. In the Mack truck for instance, gear ratios can be made from 6½ to 1, to 14 to 1, without altering anything but the sprockets and perhaps adding or taking out a link or two of the chain. In this truck the teeth of the front sprockets are hardened and the sprockets are cut from blanks of open-hearth steel, for these reasons greater wearing qualities are claimed. When sprockets are worn until they are hooked, that is to say, until the true form has been destroyed and the teeth are slightly hooked at the ends of the driving side, they will cling to the chain as it leaves the sprocket, and are then pulled away violently, although it is claimed that this can be remedied by some forms of sprocket design. The chain drive is superior to the worm drive under heavy load and torque and only slightly less efficient than the worm under high speeds and light loads.

The Internal-gear Driven Axle.—In the internal-gear driven axle first a bevel reduction is provided between the propeller shaft and the rear axle, as in most passenger cars, and to obtain a further reduction at the end of the live axle, near the wheel, or in the hub, small spur-gears mesh with internally cut gears attached to the wheel.

The advantages claimed for the internal gear driven axle are:— (See W. V. Torbensen, *S.A.E. Transaction*, 1914, Part 1.)

1. Great strength, since it permits the use of a solid, one-piece load carrying member; it is usually a drop forging of I-section, or a round axle made from bar stock. Sometimes this member is flattened and shaped in the center to partly encompass the differential housing; in some designs the live axle is within the load carrying member as shown in the illustrations.

2. Lightness. This axle can be made lighter than the worm gear type, since the jackshaft and the differential run at a lower torque for a given engine power than in the worm drive axle, the great speed reduction being made at the wheel, while the reduction at the bevel gears in the center of the axle is usually less than two to one.

3. The internal spur-gear and pinion are mounted on fixed centers and require no adjustment.

4. The cost is low, while the efficiency is high and fairly constant at all loads.

One of the serious problems with most of the internal gear driven axles is the proper lubrication of all the parts of the drive, although it is claimed by its sponsors, that this drive can run with very little lubrication. The large diameter at the ring gear gives plenty of opportunity for the grease to leak out around the edges of the gears and upon the brake. However, some types run in a constant bath of oil, and the joint is not close to the large diameter of the ring gear, but nearer the center.

Another objection against the internal gear is that it is more difficult to provide with double internal brakes on the wheel. Even where the internal brake is used, considerable difficulty is experienced with the rise in temperature due to the application of the brakes, which may cause the lubricant to melt or burn. At present the tendency is to use internal rather than external brakes, as the latter form is exposed to dust and grit. The internal gear, as usually constructed, cannot be made in the semi-floating construction, thereby eliminating the use of this design where the bearing pressures can be considerably reduced over those of the full-floating type.

Opponents of the internal gear axle further claim that it has fundamental defects. In most of the internal gear types the load-carrying member is the dead axle, but in some, a central live member is inside the dead member, one or more extra idler

gears being inserted between the jackshaft pinions and the internal gear rings in the wheel hub. Sometimes the internal gear forms a part of the brake drum, at other times it is a separate unit. In all internal gear axles the internal gear is carried by the wheel, or else in the wheel hub between bearings, and all wheels will develop a certain amount of play, since they have to take the road shocks. They will therefore not continue to run perfectly, and the gears will not maintain their exact relation. If the gears run out of alignment with each other, instead of on a line contact, the contact will take place only on a point, under which condition gears cannot wear well or run silently, nor transmit the power efficiently, for the slight play in the wheel will cause it to rock to-and-fro slightly.

In most internal gear axles, the wheels are mounted on some form of dead axle while the jackshaft, running parallel, is either in front or behind it, and encased in a separate housing, whose only function is to retain the live members. In some types, the jackshaft is attached to the axle at the center, and in others only at the end. If the dead axle and the live axle do not retain their parallelism there will be an additional tendency to throw the gears out of alignment. The torque reaction tends to revolve the jackshaft about the dead axle and is restrained from so doing only by the security of the end fastenings and by the rigidity of the dead axle against twisting. If the torque were always equal at both ends of the jack shaft, the twisting moment would be balanced, but since the traction of the two wheels is not absolutely equal, varying greatly at times, the torque stress on the axle is not equal on both ends. It is therefore claimed by opponents that this inequality of torque upsets the absolute parallelism of the two members, which will have a tendency to throw the pinions out of alignment with the internal gears, and also to subject the bearings of the differential to added stresses. In some constructions the load is carried on the dead axle, the jackshaft being perfectly free, not centrally supported, and is therefore not exposed to deflection when the axle is deflected, and this will have a tendency to pull the gears out of line and cramp the pinion bearings. Sometimes means are provided to take care of the deflection by providing a universal joint in the jackshaft, but in this country such a construction is not used to the author's knowledge; in the axle of the De Dion Bouton Co., and the Rochet-Schneider Co., of France, it is used in order to obtain

quiet operation. This, on the other hand, introduces a more complicated mechanism and a larger number of parts.

The type which has a separate dead axle weighs more for a given strength, since the larger outside dimensions of a hollow tube, offer more strength with less weight, hence, for the same factor of safety, a solid axle is heavier than a tubular construction.

However, against the disadvantages enumerated is the fact that about 17 per cent of the models of trucks built in this country employ the internal gear drive and among its adherents are some of the well known companies.

The Worm-drive Axle.—(See John Younger, S.A.E. *Transactions*, 1914, Vol. I from which a portion of this chapter has been taken.) John Dennis was the first to adopt the worm gear to motor truck rear axle drives on a 3½-ton omnibus. The worm and worm gear are usually assembled in a unit and then attached to the axle housing. This type of construction lends itself very readily to accurate machine work. It is very rigid and prevents the road shocks or stresses, other than those coming through the driving axle, from disturbing the alignment of the worm gear. The housing for the live axle is tubular and very solid. The thrust is often taken up by the conventional type of radius rod; universal joints at both ends are advantageous in eliminating undue stresses. A later development is to take the thrust and the torque through the main springs (the Hotchkiss drive) and this is now successfully used by a number of concerns. Provisions should be made for preventing the oil in the center of the housing from running out through the wheel hubs. Oil retainers are sometimes used on both the axle shafts and the wheel hubs for this purpose. A vent pipe in the rear axle housing will also assist in overcoming oil leaks. It is claimed that the worm gear efficiency will average about 95 per cent, and that a truck equipped with a high ratio will cost quiet freely.

The great advantages claimed for the worm gear axle are the perfect mechanical enclosure of the final drive which is running in a bath of oil and its practically noiseless operation. Even the wheel bearings are lubricated from the center housing, and when proper provisions are made there is no danger of the lubricant saturating the brakes. When the housing is filled with suitable oil, the latter will last up to 5,000 miles. Due to the

perfect enclosure, dirt and grit are positively prevented from reaching the mechanism. Worm gear axles are frequently equipped with double internal brakes, which affords excellent protection against dirt and grit. It is claimed that wheel brakes are more satisfactory than propeller shaft brakes, for with the latter the stresses in the drive shafts and the gears are considerably greater than when wheel brakes are employed, for the torque arising from the locking of the wheels together with their momentum of the large capacity trucks, is much higher than that of the engine. Other advantages claimed for the worm gear are:—(1) its simplicity, it having the fewest number of parts; (2) it is possible to obtain the desired reduction with but two pieces, the worm and the worm wheel; (3) it is possible to obtain an almost straight drive from the engine to the rear axle; (4) its durability is very great.

It is better to have the weight in the center of the axle than at the axle ends, for when one wheel rises or hits an obstruction and is thrust upward, the upward acceleration in the center of the axle will be only one-half that at the wheel, hence the stresses induced will be smaller, and the action will be more like that of a smaller unsprung weight than if the weight were at the wheels. The life of the worm gear is well over 50,000 miles, if attention is paid to lubrication.

Opponents of the worm gear claim: (1) It is a highly efficient power transmission gear only under certain conditions. Under a heavy torque, a worm gear has a low efficiency, and a comparatively large amount of power is consumed within the gear itself. At high speed, under light load, a worm is highly efficient. (2) In coasting there is a certain amount of friction in the worm and gear. Tests made by the International Motor Co. showed that trucks equipped with a worm drive did not coast as well as other types; hence it is claimed, that even though the worm may be more efficient than the chain under certain conditions, its average efficiency, throughout the range of conditions encountered in service, is lower.

The most fragile part of the worm drive is the worm thrust-bearing, which takes the enormous thrust exerted by the worm. In some axles, ball bearings are provided at this point; others are equipped with taper roller bearings. The trouble of this bearing is chiefly due to the fact that the worm must be carried on the top to obtain sufficient road clearance, and the worm bearings

above can be lubricated only by the oil carried up, or thrown up, as the worm wheel rotates. When this runs at low speed, sufficient oil is not always carried to the top.[1]

To provide sufficient lubrication for the thrust and radial bearings at the rear, special means, like oil-troughs, are sometimes provided inside the worm wheel housing, to catch the oil and lead it to the bearings. The worm drive is more expensive than most other forms, but it is claimed that the maintenance cost is considerably less. When the pressure on the worm is not too high, the gears properly cut, and the lubrication adequate, the film of oil is not squeezed out from between the surfaces, and there is very little friction. However when the pressure is excessive, and the oil is squeezed out, considerable friction exists and damage may result in a short time.

Another reason for rear bearing failure is that the necessary play in the worm takes the form of a blow on the thrust-bearing whenever the clutch is thrown in or speed changes are made, and while taper roller bearings and ball bearings can withstand high pressures, they cannot withstand impact very well. Trouble to thrust-bearings may also be caused by improper clearances, which, under excessive strain or deflection, cause an increase in temperature with a consequent elongation of the worm.

In the worm drive, the torque induces pressure on comparatively small surfaces of the teeth, which rub, one on the other, while with toothed gears or chains and sprockets, there is more of a rolling contact than pure friction. Some manufacturers claim that torque arms and radius rods impose additional stresses on the worm, therefore they advocate the Hotchkiss drive. In this, a cushioning effect is imparted to the drive as the springs permit the axle to rock back slightly in starting, thereby reducing the pressure on the worm tooth when the car is first started and the torque highest. It is also claimed that it permits the worm and the wheel to oscillate, thereby working more oil between the surfaces than is possible with a rigid drive. On the other hand opponents of the Hotchkiss drive in connection with worm gears, claim that they must put extra weight into the springs when the latter are used to perform the functions of the torque and radius rods, and wherever satisfactory results are obtained with the Hotchkiss drive the springs must have a greater factor of safety.

[1]See H. D. CHURCH, *S. A. E. Transactions*, 1916, Part II.

It is further claimed that in order to obtain a successful Hotchkiss drive it is necessary to have fairly flat rear springs.

The author believes that the Hotchkiss drive, when properly designed gives satisfaction, even though the springs are relied upon to perform more severe functions.

While many objections are brought forward against the worm drive, about 75 per cent of all the truck models manufactured in the United States use this type of final drive.

The Double-reduction Drive.—In the double-reduction rear-axle two reductions take place between the propeller-shaft and the live axle. The first reduction is by a pair of bevel gears and the second by a pair of spur gears, or vice versa. Instead of straight-bevel or spur-gear teeth, helical teeth can be employed.

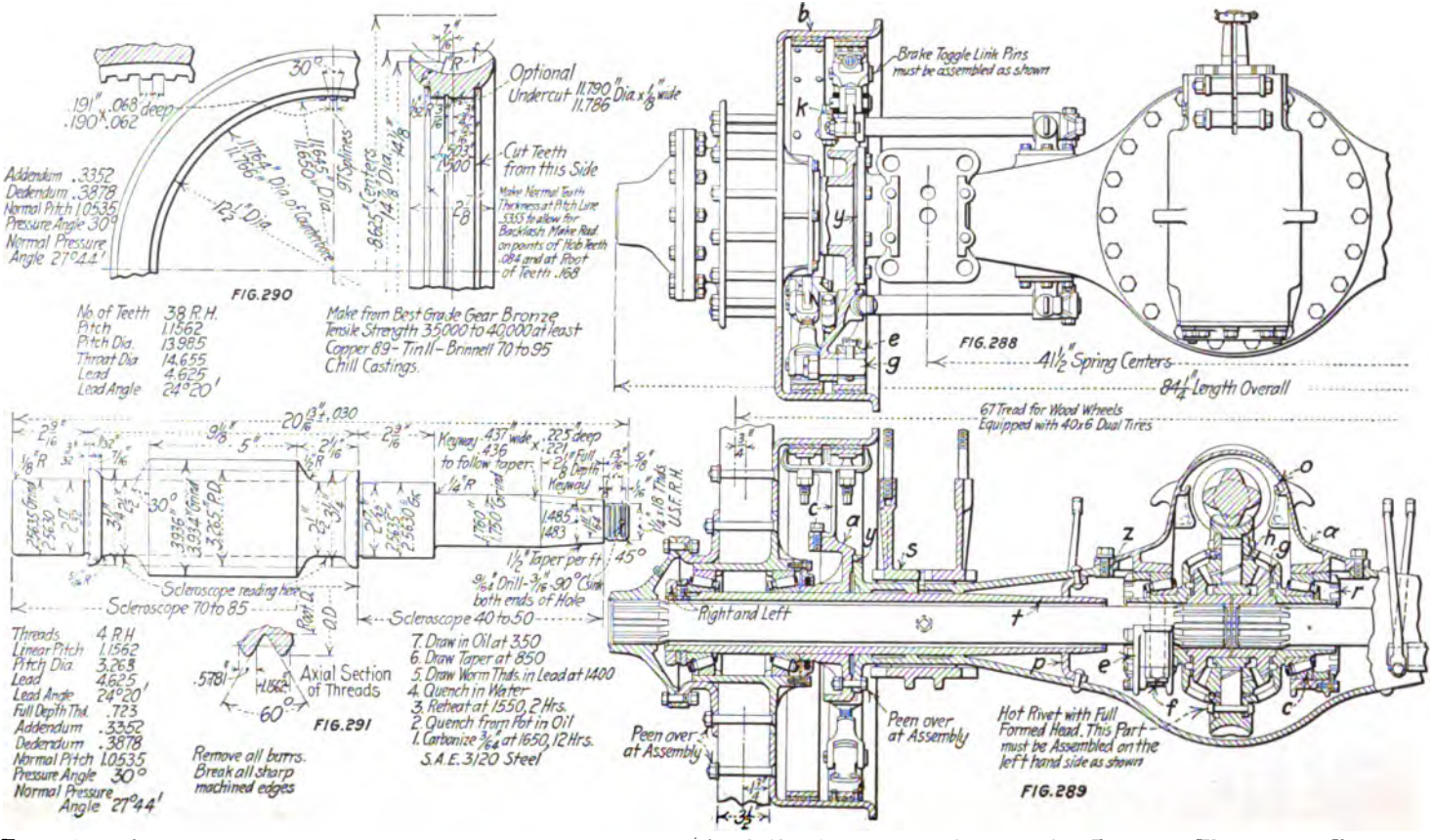
The advantages claimed for this type are (*a*) simplicity of construction, in that only comparatively small bevel and spur gears are employed, which lend themselves easily to quantity production; (*b*) all the gears are perfectly enclosed and protected from dust and grit, and run in a bath of oil; (*c*) its average efficiency is as high as that of any other drive and is substantially constant under all speeds and loads; and (*d*) the gears will always remain in alignment and are silent in running.

The disadvantages cited against this drive are (*a*) an increased number of parts over the wormgear drive; (*b*) it is heavier than the internal-gear and the chain drive, since the live axle has to carry the entire torque; and (*c*) it is more costly to manufacture.

Practical Examples of Truck Rear Axles. *Worm Drives.*—Figure 288 is a plan view of the Class B rear axle.[1] The housing is of the Timken type made of pressed steel with a square section from the center bowl outward.

Figure 289 shows a sectional view looking from the rear. The tube *t* is pressed into place; a retaining screw being provided in addition. This tube extends to the bowl of the housing; reinforcing plate *p* is riveted to the housing and fits snugly over the end of the tube. If desired this plate may be welded to the housing. The spring seats *s* are of forged steel and made in two halves—a lower and an upper (see Fig. 292); they fit over machined surfaces of the square portion of the housing and a dowel *d* keeps them in position. The worm and worm wheel, (Fig. 293), are of the David Brown type with 8⅝ in. center distance, a pressure angle of 30°, linear pitch 1.1562 in., and lead angle of

[1]See G.W. Carlson *S.A.E. Trans.* 1918, Part I.by


Optional Undercut 11.790" Dia. x 1/8" wide 11.786
Cut Teeth from this Side
Addendum .3352
Dedendum .3878
Normal Pitch 1.0535
Pressure Angle 30°
Normal Pressure Angle 27°44'
FIG. 290
No. of Teeth 38 R.H.
Pitch 1.1562
Pitch Dia. 13.985
Throat Dia 14.655
Lead 4.625
Lead Angle 24°20'
Make from Best Grade Gear Bronze
Tensile Strength 35000 to 40000 at least
Copper 89 - Tin 11 - Brinnell 70 to 95
Chill Castings.
Brake Toggle Link Pins must be assembled as shown
FIG. 288
41 1/2" Spring Centers
84 1/4" Length Overall
67 Tread for Wood Wheels Equipped with 40x6 Dual Tires
Keyway .437" wide x .225" deep .436 to follow taper
1 1/2" Taper per ft
Sclerescope reading here
Sclerescope 70 to 85
Sclerescope 40 to 50
Right and Left
Peen over at Assembly
Peen over at Assembly
Hot Rivet with Full Formed Head. This Part must be Assembled on the left hand side as shown
FIG. 289
Threads 4 R.H.
Linear Pitch 1.1562
Pitch Dia. 3.268
Lead 4.625
Lead Angle 24°20'
Full Depth Thd. .723
Addendum .3352
Dedendum .3878
Normal Pitch 1.0535
Pressure Angle 30°
Normal Pressure Angle 27°44'
Axial Section of Threads
FIG. 291
Remove all burrs. Break all sharp machined edges
7. Draw in Oil at 350
6. Draw Taper at 850
5. Draw Worm Thds. in Lead at 1400
4. Quench in Water
3. Reheat at 1550, 2 Hrs.
2. Quench from Hot in Oil
1. Carbonize 3/64" at 1650, 12 Hrs.
S.A.E. 3120 Steel

Figs. 288 and 289.—Rear view and cross-section of worm drive rear axle of Class B, U. S. A. military truck. Fig. 290.—Worm gear. Fig. 291.—Worm and shaft.

24°–20′. *w* shows the worm gear, *v* the worm. The worm gear is mounted on the differential by means of 97 corrugations or splines $\frac{3}{16} \times \frac{1}{16}$ in. for a length of $\frac{15}{16}$ in.; see Fig. 290 which gives the details of the worm gear. It is made of the best grade gear bronze and is a chilled casting; the material has a minimum tensile strength of from 35,000 to 40,000 lb. All details are noted in the drawing. It is secured against side thrust by a ring shield with 16 rivets, against a shoulder on the opposite side.

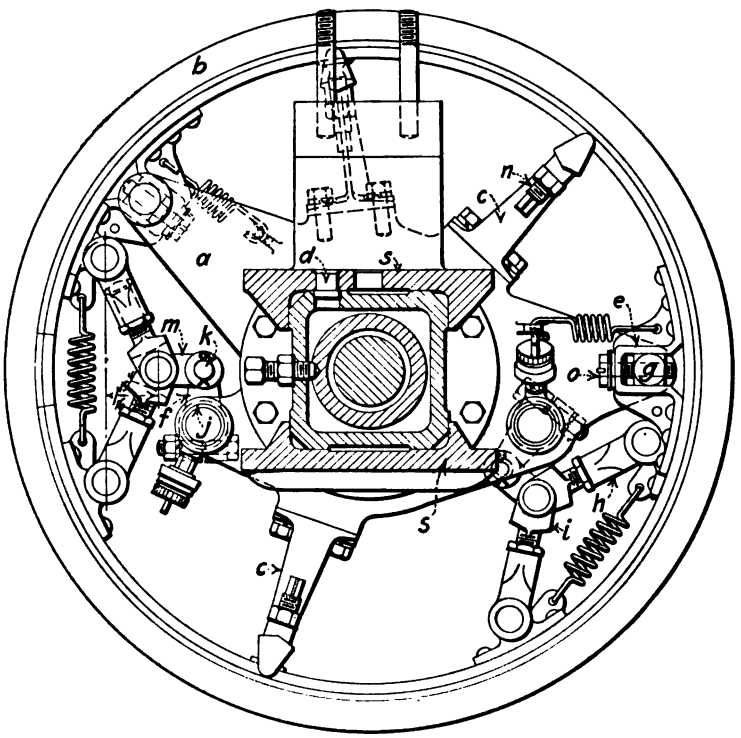


FIG. 292.—Section through spring seat (Class B truck).

Figure 291 is a detail of the worm; this is made of 3120 S.A.E. chrome-nickel steel, hardened and ground.

The differential, and the worm and gear, are mounted on the differential carrier (see Fig. 294), piloted at the top of the housing as seen at *z* in the assembly, (Fig. 289). This carrier (*a* in the assembly) is a malleable iron casting. At the center the metal thickness is $\frac{3}{16}$ in., being increased where added strength is

required, as for instance, at the ends wherein the bearings are supported; the ribs, are 7/16 in.; at *o* are oil grooves to lead the oil to the worm bearings; at *b* two lugs for the attachment of a rope, if necessary, for the purpose of lifting the carrier out of the housing. The means used for holding the worm bearings in place are seen from Fig. 293. The gear adjustment is made by slotted and threaded rings (Fig. 295) back of the bearing differential cups, as shown by *r* in the assembly.

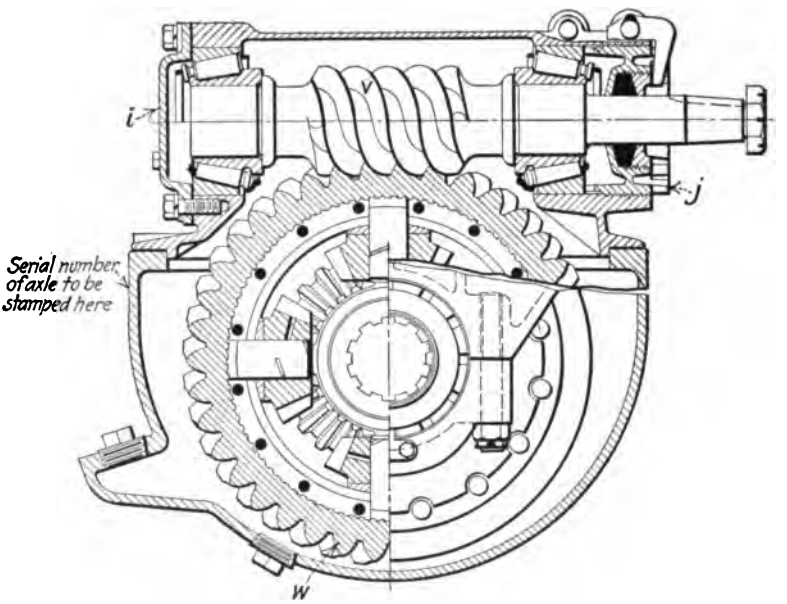


FIG. 293.—Worm and worm gear (Class B truck).

The roller bearing cup *c* is clamped to the upper main differential carrier by cap *e* which is held by nickel steel studs *f*, 1 in. in diameter. The entire unit can be removed from the housing without disturbing any adjustments.

Throughout the axle the various parts have been so designed, wherever possible, as to permit assembling and disassembling of the following units: (1) The shaft and drive plates; (2) the wheel, drums, bearings and thrust washer. (3) brake, brake anchor and bands, levers and shafts, and (4) the differential carrier and its parts. The six roller bearings, in the wheels and at each side of the differential, are of the same size and thus interchangeable.

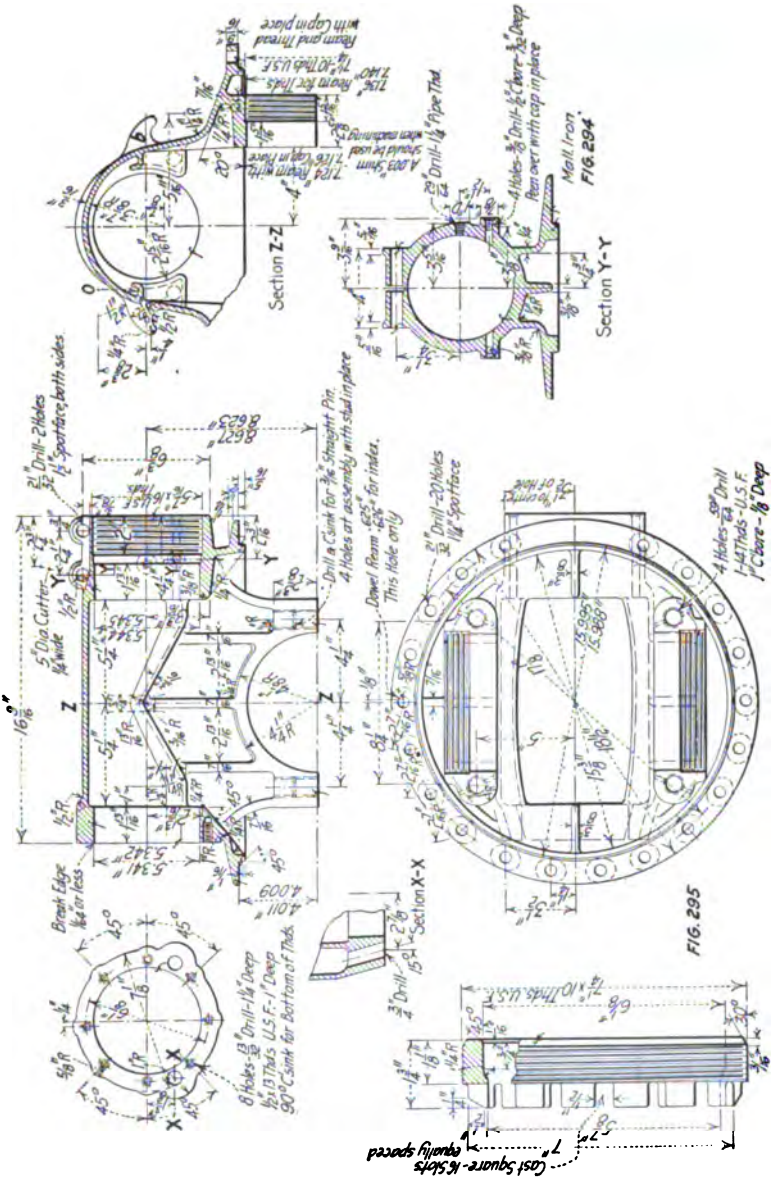


FIG. 294.—Differential carrier. FIG. 295.—Differential bearing adjusting ring (Class B rear axle).

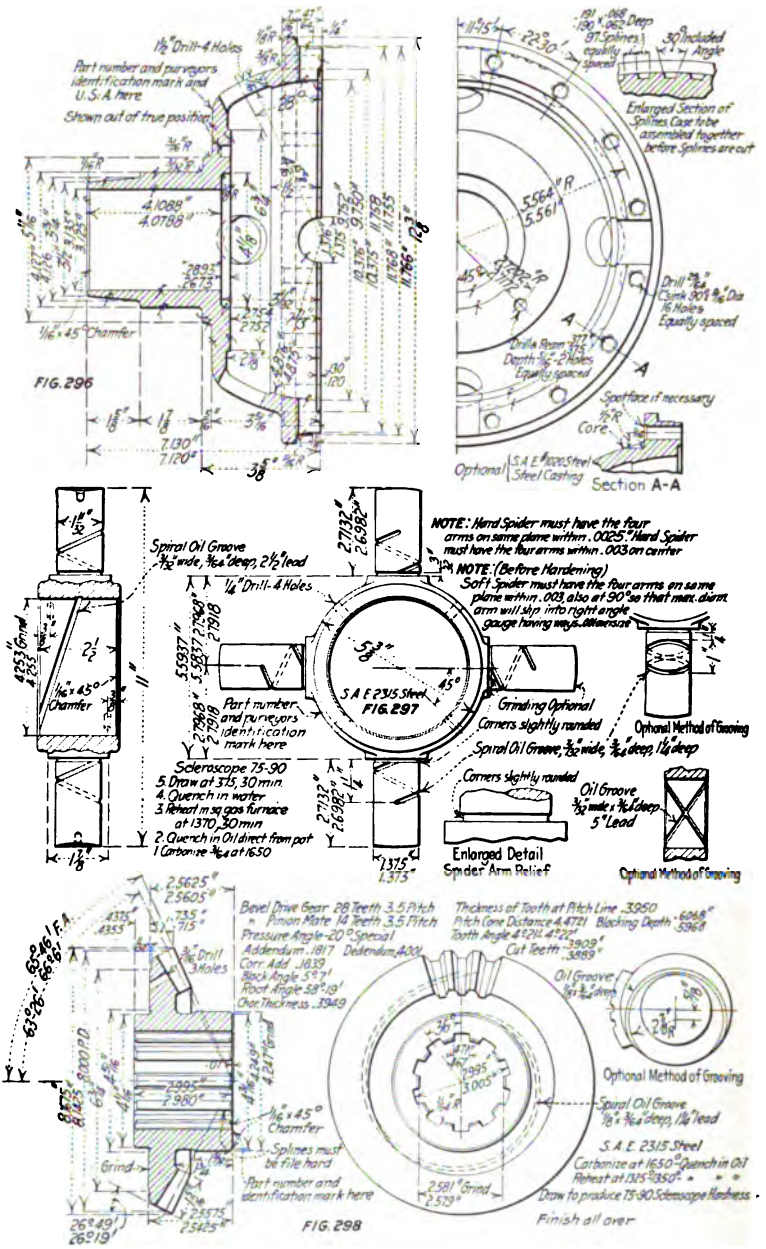


Fig. 296.—Differential case. Fig. 297.—Differential spider. Fig. 298.—Differential bevel side gear (Class B rear axle).

Figure 296 shows one-half of the differential case which may be either a steel casting or a stamping of 1020 S.A.E. steel. The upper view shows the corrugations or splines, for the worm wheel, before described. Figure 297 is the differential spider made of 2315 S.A.E. steel.

Figure 298 is the differential bevel side gear and Fig. 299 the bevel side pinion which revolves on the spider arm having a diameter of 1.373 to 1.375 in., the bore of the pinion being from 1.377 to 1.379 in., showing the minimum clearance to be .002 in. and the maximum .006 in., for a running fit. Note the oil hole through the pinion and the groove at *i* where it is seated on a steel thrust washer *h*, a detail of which is given in Fig. 301.

Figure 300 shows the hardened thrust washer, located between the differential bevel side gear and the differential case, shown at *g* in the assembly.

Figure 302 is the rear bearing cover and Fig. 303 the gasket for the same (see *i*, Fig. 293).

Figure 304 is the adjusting ring at the front end of the worm shaft. It is screwed into the differential carrier, as seen from the worm gear assembly, with a felt packing ring on the inside.

Figure 305 is a detail of the housing, a steel stamping; it is made of ⅜-in. stock, reinforced as shown. This is an optional construction, slightly differing from that shown in the assembly, or in the next figure. In this drawing the axle tubes are not shown. Figure 306 is a detail of the housing when made in a steel casting (the government having given manufacturers the choice of making it of pressed steel or a steel casting) with the section at various points, the thickness of the metal varying from 5/16 in. at the center to ½ in. near the end, as the stress increases toward the end. Figure 307 is a detail of the right hand sleeve which is inserted from the end of the housing (when made of pressed steel) as seen from Fig. 289. The sleeves are made of nickel-chromium steel, the metal thickness being about ½ in. in the center.

Figure 308 is the axle shaft, or drive shaft, made of chrome-nickel steel with high carbon contents, (S.A.E. 3140). As seen, the diameter is 2½ in. at its weakest point. The axle is of the full-floating type, hence the radial as well as the lateral loads on the wheel are carried by tapered roller bearings, the shaft transmitting only the torque, *i.e.*, it is only under torsional strain. Figure 309 is the drive plate which is shrunk on the end

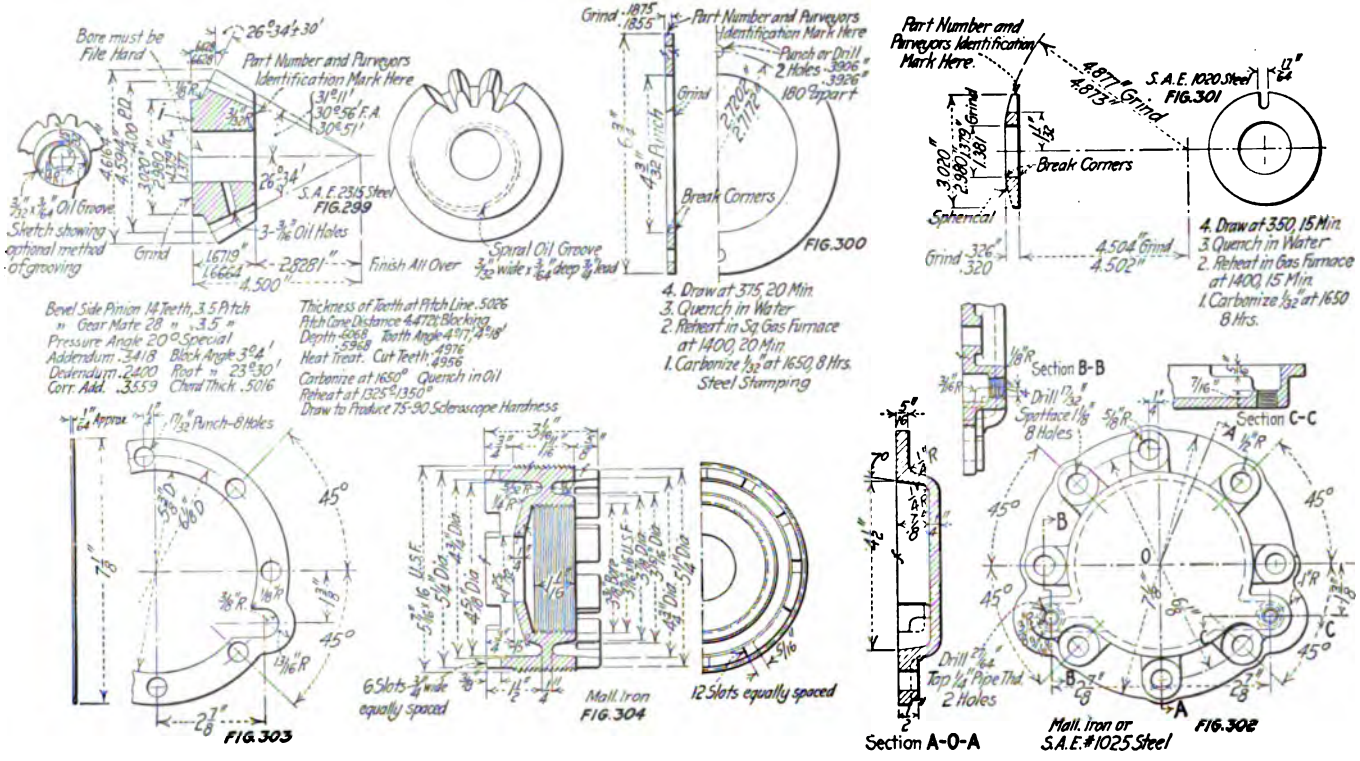

Bore must be File Hard
Part Number and Purveyors Identification Mark Here
Oil Groove
Sketch showing optional method of grooving
Grind
3-3/16" Oil Holes
Finish All Over
S.A.E. 2315 Steel
FIG. 299
Spiral Oil Groove
Bevel Side Pinion 14 Teeth, 3.5 Pitch
Gear Mate 28 " 3.5 "
Pressure Angle 20° Special
Addendum .3418 Blank Angle 39° 4'
Dedendum .2400 Root " 23° 30'
Corr. Add. .3559 Chord Thick .5016
Thickness of Tooth at Pitch Line .5026
Pitch Cone Distance 4.4721 Blocking Depth .4068 Tooth Angle 4° 17' 28"
Heat Treat. Cut Teeth: .4976 .4956
Carbonize at 1650° Quench in Oil
Reheat at 1325°/1350°
Draw to Produce 75-90 Scleroscope Hardness
Grind
Part Number and Purveyors Identification Mark Here
Punch or Drill 2 Holes
Break Corners
FIG. 300
4. Draw at 375, 20 Min.
3. Quench in Water
2. Reheat in Sq. Gas Furnace at 1400, 20 Min.
1. Carbonize 1/32" at 1650, 8 Hrs.
Steel Stamping
Part Number and Purveyors Identification Mark Here.
S.A.E. 1020 Steel
FIG. 301
Break Corners
Spherical
4. Draw at 350, 15 Min.
3. Quench in Water
2. Reheat in Gas Furnace at 1400, 15 Min.
1. Carbonize 1/32" at 1650
8 Hrs.
Section B-B
Drill 17/32" Spotface 1 1/8" 8 Holes
Section C-C
Section A-O-A
Drill 27/64" Tap 1/4" Pipe Thd. 2 Holes
Mall. Iron or S.A.E. #1025 Steel
FIG. 302
45°
FIG. 303
6 Slots 3/4" wide equally spaced
12 Slots equally spaced
Mall. Iron
FIG. 304

Fig. 299.—Differential bevel side pinion. Figs. 300 and 301.—Thrust washers. Fig. 302.—Bearing cover. Fig. 303.—Gasket. Fig. 304.—Adjusting ring.

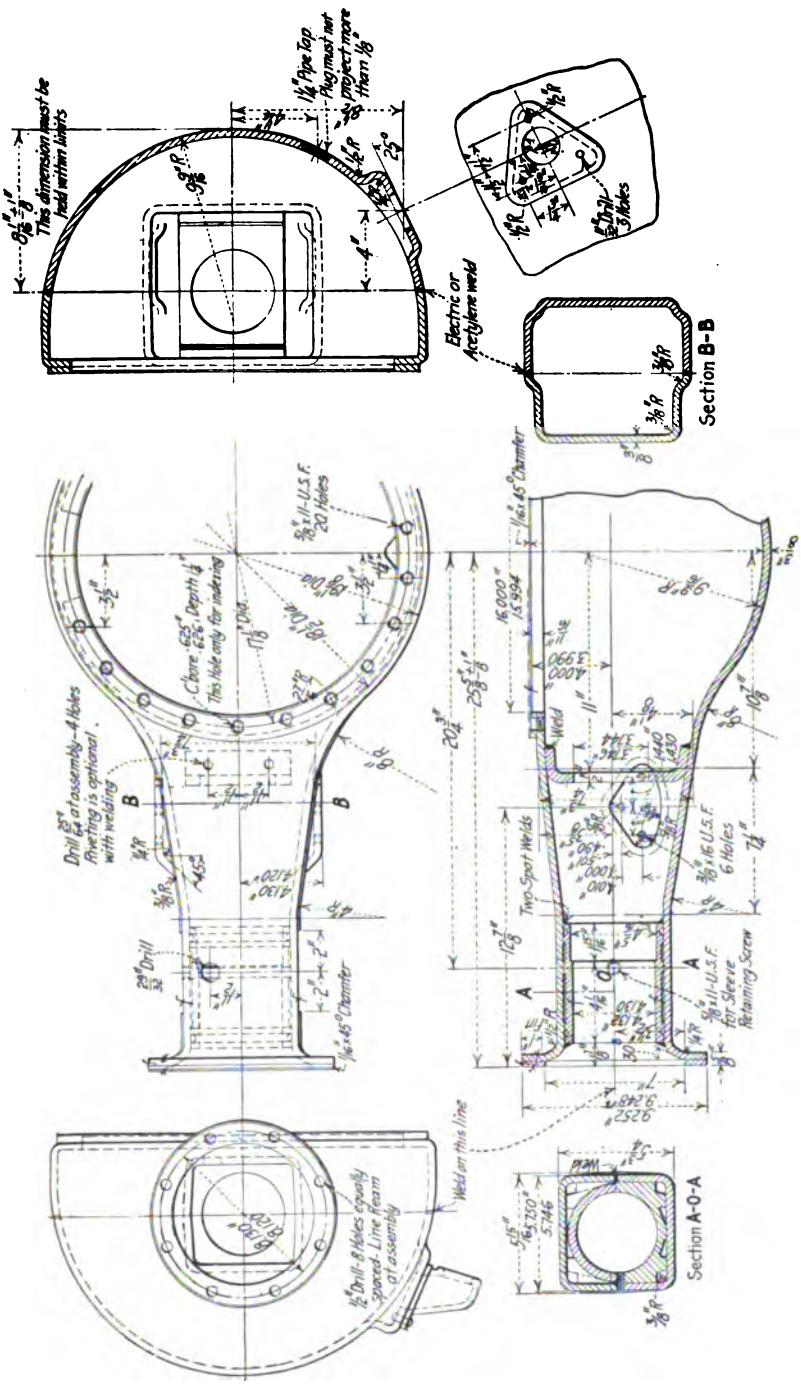

This dimension must be held within limits
1/4" Pipe Top. Plug must not project more than 1/8"
Electric or Acetylene weld
Section B-B
Section A-O-A
Weld on this line
Two-Spot Welds
Weld
Drill 25/64" at assembly—4 Holes Riveting is optional with welding
Clearance .625" Depth 1/8" This hole only for indexing
45° Chamfer
Drill 8 Holes equally Spaced-Line Ream at assembly

FIG. 305.—Rear axle housing; steel stamping (Class B, rear axle).

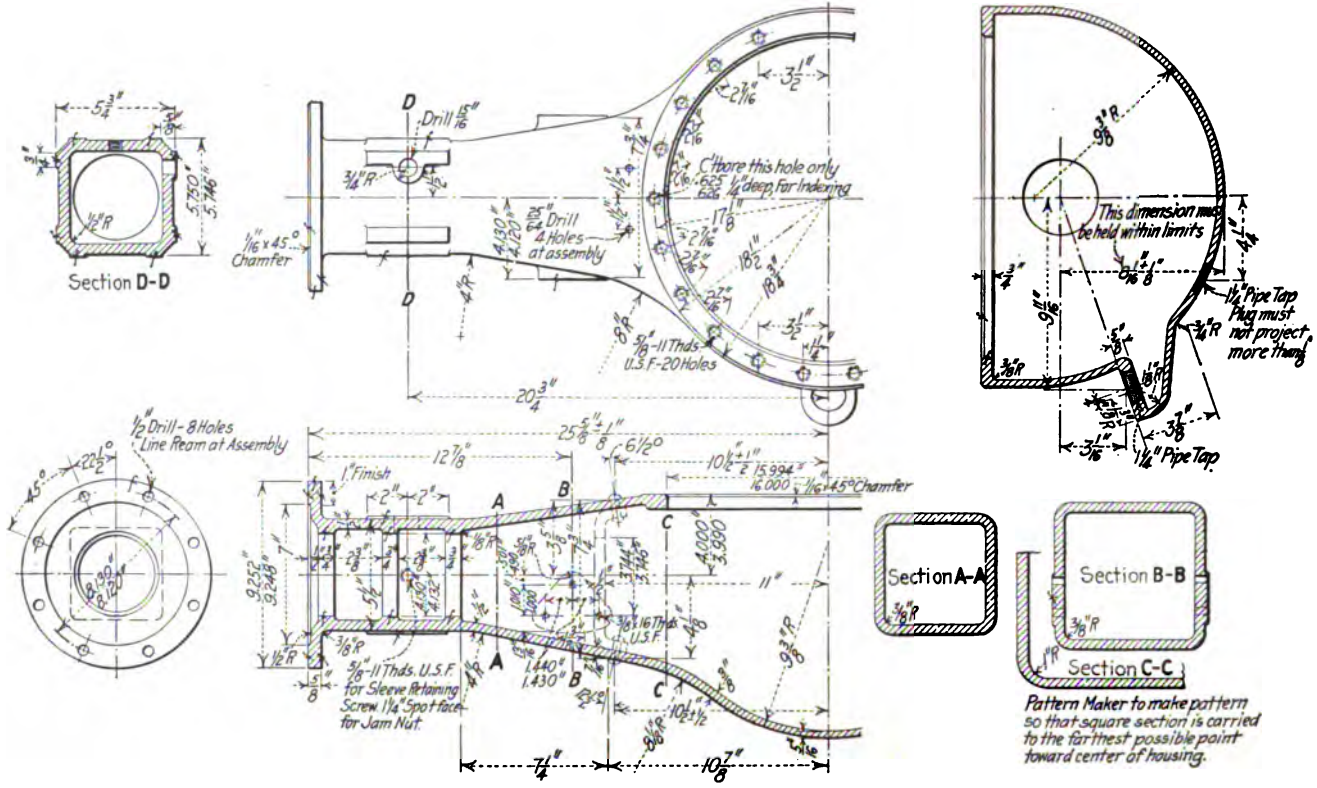

Section D-D
Drill
Chamfer
Drill 4 Holes at assembly
C'bore this hole only
Pipe Tap
Plug must not project more than
This dimension must be held within limits
Pipe Tap
Drill - 8 Holes
Line Ream at Assembly
Finish
Chamfer
Section A-A
Section B-B
Section C-C
Thds. U.S.F.
for Sleeve Retaining Screw. Spot face for Jam Nut.
Pattern Maker to make pattern so that square section is carried to the farthest possible point toward center of housing.

FIG. 306.—Rear axle housing; steel casting (Class B, rear axle).

of the drive shaft; it is made from .010 to .020 in. smaller than the latter (over the outside of the splines) to allow for shrinkage. Figure 310 is the hub flange to which the drive plate is attached; the 14 holes on the larger flange are utilized for its attachment to the wheel.

Brake Mechanism.—The brake mechanism may be seen from the assembly, (Fig. 292), where *a* is the brake anchor; Fig. 313 shows a detail of the same (right hand) made of S.A.E. 1025 steel. It is seated on the axle tube as seen in Fig. 289, the brake band support *c* (Fig. 311) being attached to it by means of bolts and lock nuts, as shown in Figs. 289 and 292. Attention is drawn to the various sections of the brake anchor and the brake band supports to obtain strength and rigidity with minimum weight. The brake band support may be either a forging, a malleable casting or a steel casting, the government having given manufacturers the choice to make it of either of these. *b* is the brake drum on the inside of which two brake bands are situated side by side, which are operated by means of a toggle mechanism as seen in Fig. 292. Figure 312 is a detail of the brake drum, a steel stamping, having a minimum thickness of $\frac{5}{16}$ in. after being finished on the inside. The stamping may be made either of $\frac{3}{8}$- or $\frac{7}{16}$-in. stock. The entire width of the brake drum is 7 in., and the width of each brake band $2\frac{1}{4}$ in.

e,e, denote the brake band anchor brackets which are riveted to the brake bands; they are either steel forgings or steel castings. A detail of one of these is given in Fig. 314. Figure 315 is the brake anchor pin, the square length of which is located in the slot of the brake band anchor bracket just described. The location of *g* may be seen from Figs. 292 and 288. These anchor pins are attached to the brake anchor at *z* (see Fig. 313). Figure 316 is the brake expansion lever, shown at *f* in the assembly, which is serrated for purposes of adjustment on the brake expansion lever shaft. Figure 317 is bushing for this shaft made of S.A.E. 26 bronze; it is pressed into the brake anchor at *w*. The bushings are given a press fit of from .002 to .005 in. The brake expansion lever pin, which is attached to the upper end of the brake expansion lever is shown in Fig. 318 (marked *k* in the assembly); this operates the short link *m* which in turn actuates the toggle linkage which expands the brake band. Figure 319 is a detail of the brake toggle (female) and Fig. 320, the brake toggle (male). They are marked *h* and *i*, respectively. By

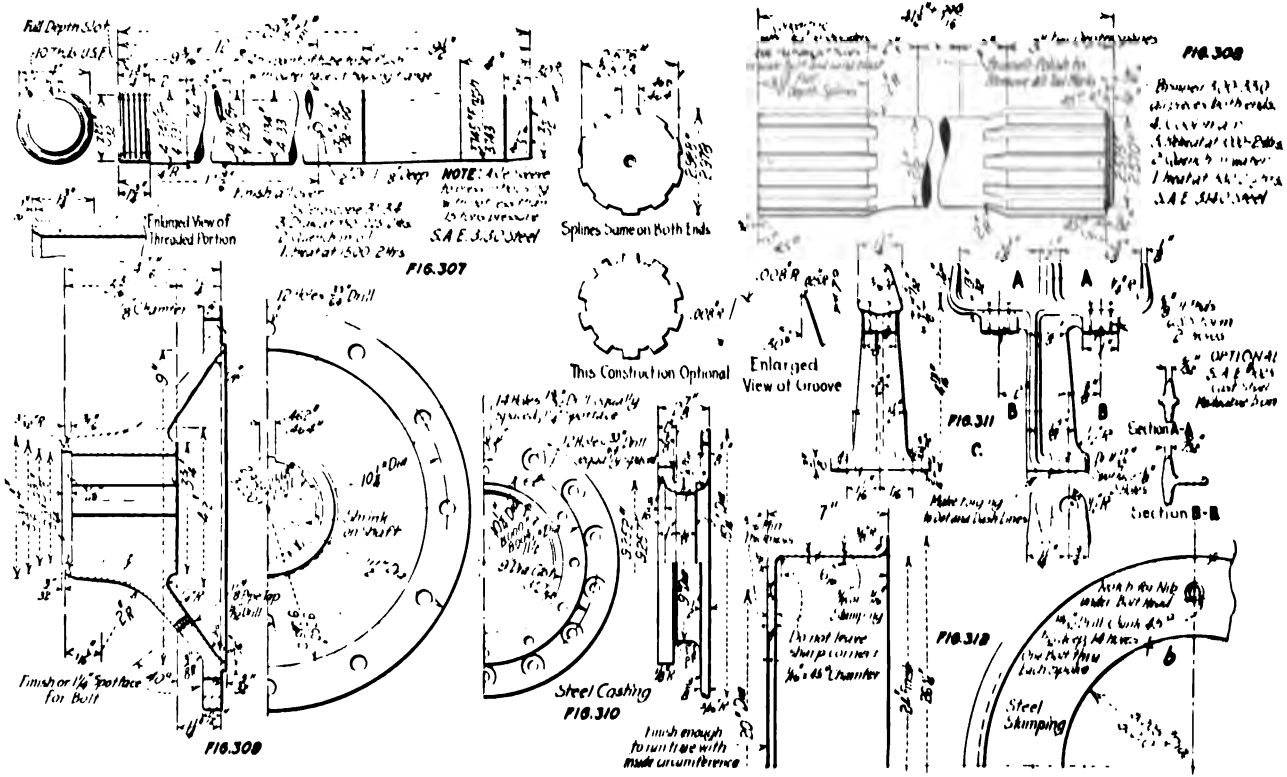


FIG. 307.—Axle housing sleeve. FIG. 308. Drive shaft. FIG. 309.- Drive plate. FIG. 310.- Hube flange. FIG. 311. Brake band support. FIG. 312. Brake drum.

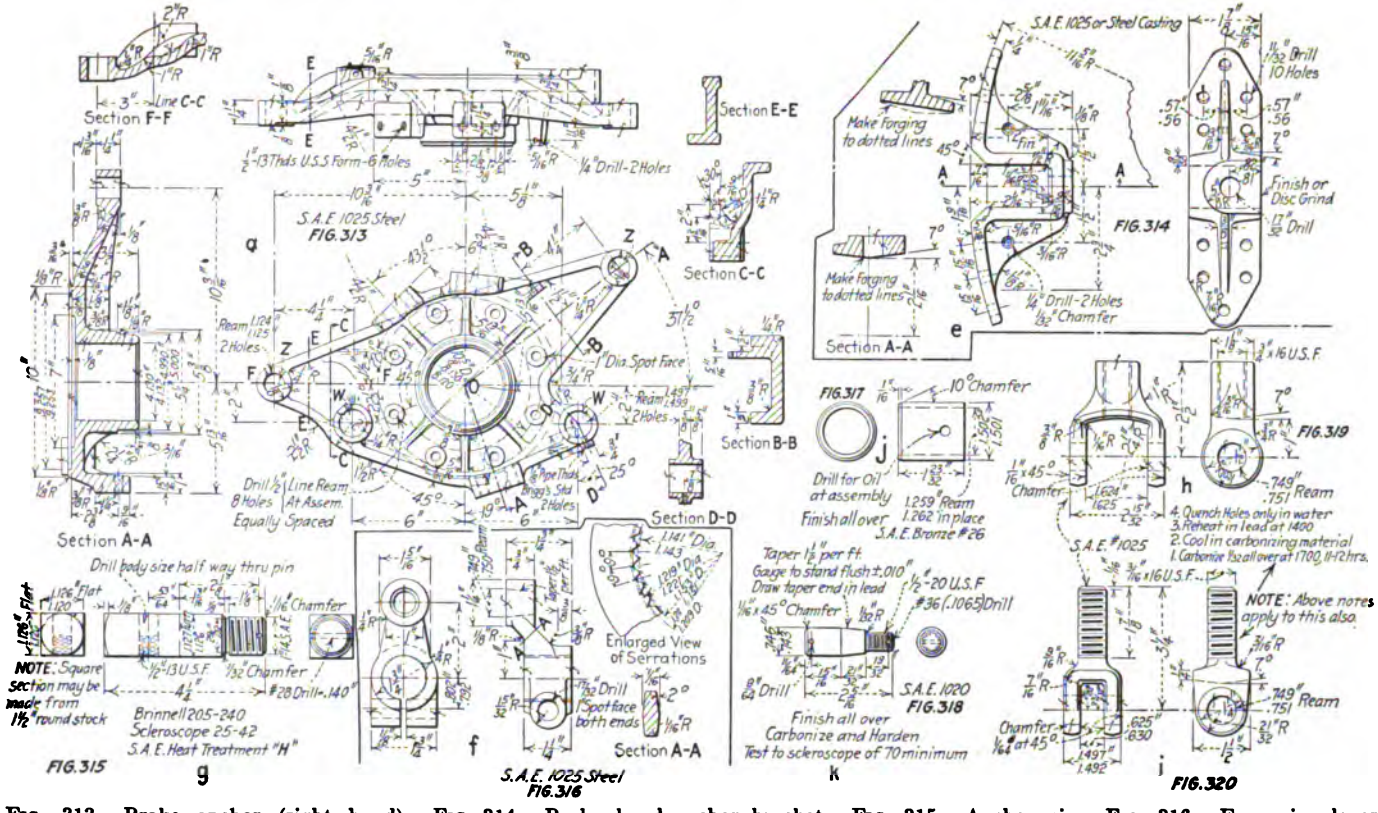


FIG. 315 g — FIG. 316 S.A.E. 1025 Steel — k — FIG. 318 S.A.E. 1020 — FIG. 320

FIG. 313.—Brake anchor (right hand). FIG. 314.—Brake band anchor bracket. FIG. 315.—Anchor pin. FIG. 316.—Expansion lever. FIG. 317.—Bushing. FIG. 318.—Expansion lever pin. FIGS. 319 and 320.—Brake toggle (male and female) (Class B, rear axle).

means of screws *n* in the brake band support, and screws *o* in the brake band anchor bracket, the distance between the brake bands and the brake drum can be adjusted. The amount of expansion of the bands may be adjusted by the screw threads of the brake toggles, male and female. It might be stated that a gasket of some approved gasket material is placed between the joint of the brake anchor and the axle housing noted at *y* in Figs. 288 and 289.

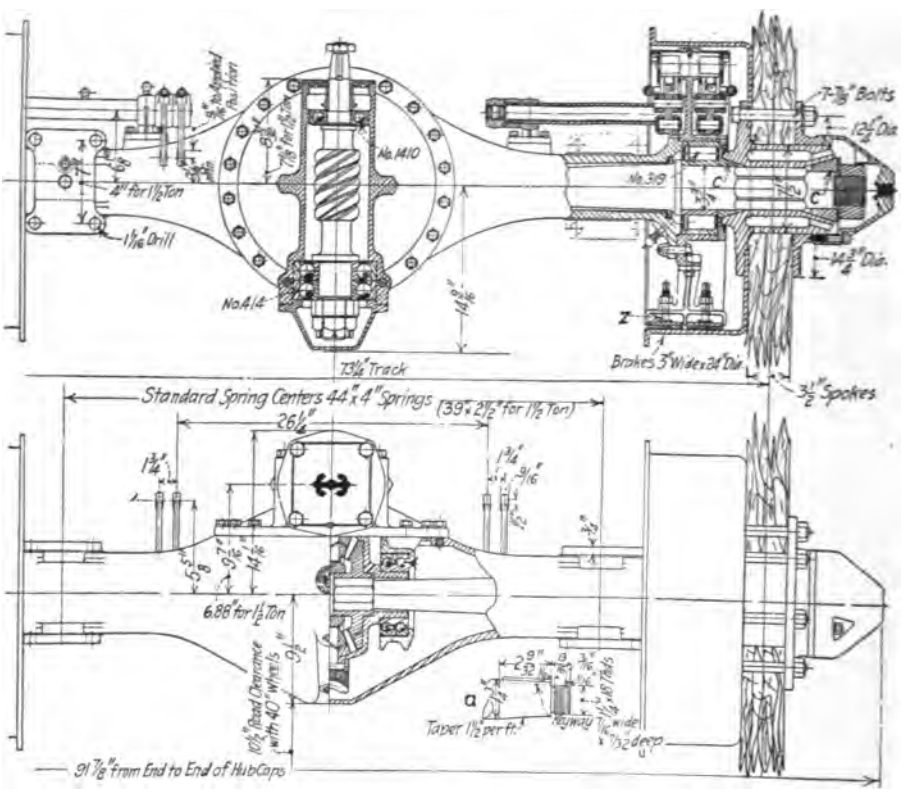


FIG. 321.—Sheldon worm drive rear axle for 5- to 6-ton trucks.

Figure 321 is the Sheldon worm drive rear axle, suitable for trucks of 5- to 6-ton capacity. The maximum load allowable on the rear tires, including weight of truck and pay load, is 18,000 lb. The weight of this rear axle complete is 2,010 lb. approximately. In all the rear axles manufactured by the Sheldon Axle & Spring Co. ball bearings are used for taking the worm

thrust as well as the radial load of the worm. The double row bearing 414 at the rear, carries the radial as well as the thrust

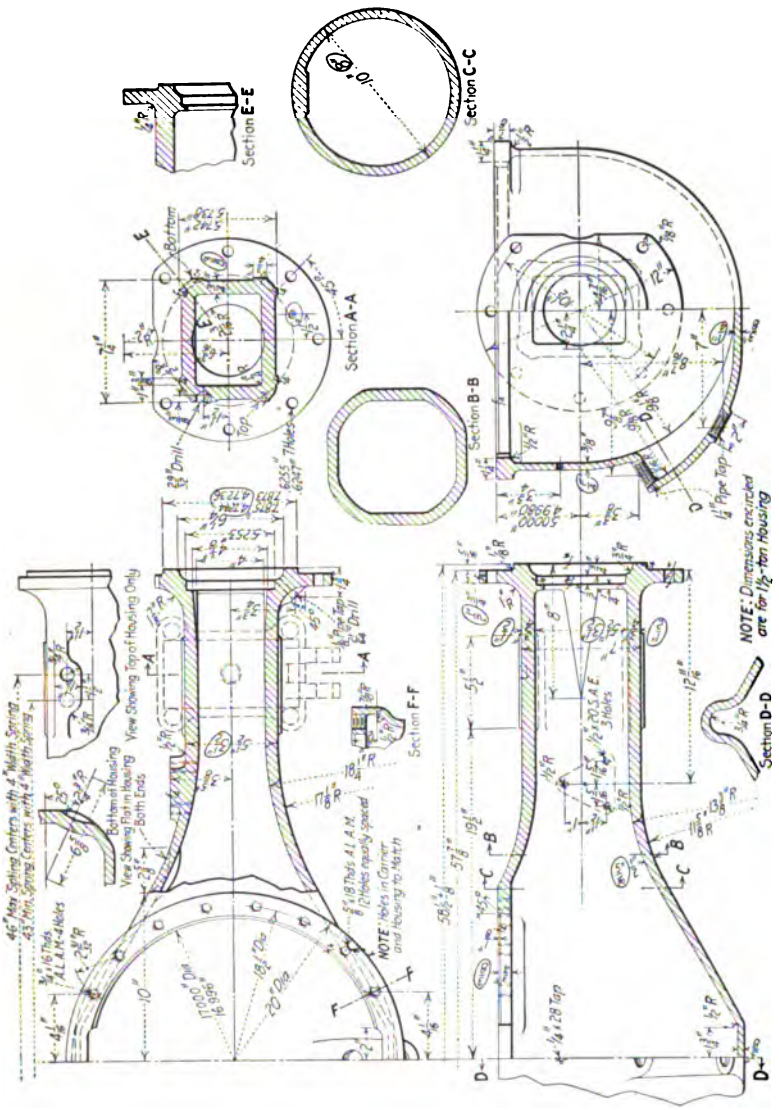


FIG. 322.—Sheldon rear axle housing (steel casting) for 5- to 6-ton truck. Dimensions encircled for 1½-ton.

load in both directions. It is claimed by the manufacturers that with ball bearings no wedging is possible under thrust,

whereas with taper roller bearings such wedging exists. The front bearing 1,410 is floating, *i.e.*, it is free to move when the worm expands, due to a rise in temperature; this may be occasioned by extra heavy work or lack of proper lubrication. No adjustments are necessary with ball bearings, and as the worm wheel and carrier are machined by very accurate jigs and fixtures, no adjustments are necessary in the first place. The semi-floating or fixed hub type of axle is employed on all Sheldon

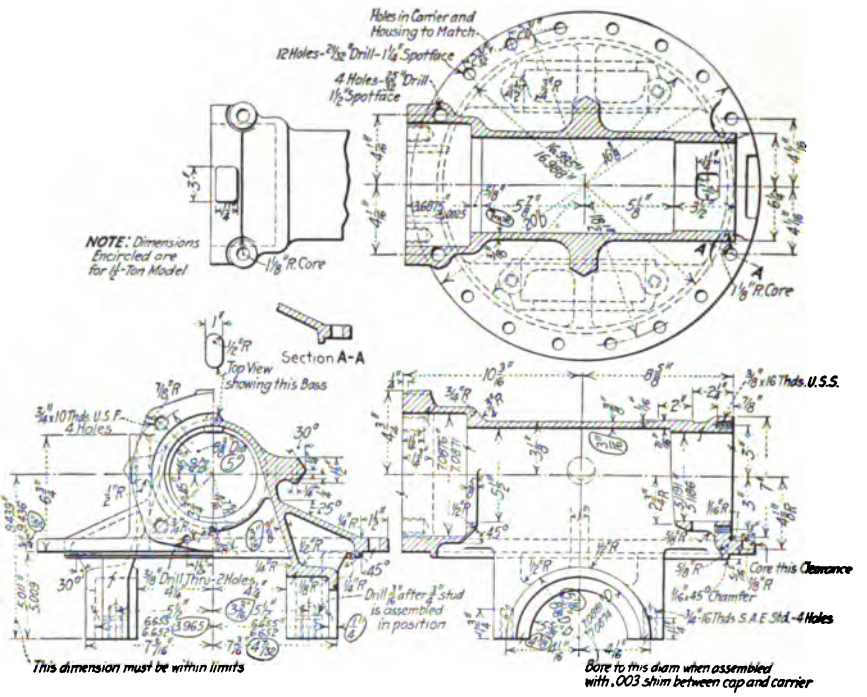

Holes in Carrier and Housing to Match
12 Holes-Drill-Spotface
4 Holes-Drill-Spotface
NOTE: Dimensions Encircled are for ½-Ton Model
Section A-A
Top View showing this Boss
This dimension must be within limits
Bore to this diam when assembled with .003 shim between cap and carrier
Core this Clearance
45° Chamfer
Thds. S.A.E. Std.-4 Holes
Thds. U.S.S.
Thds. U.S.F.-4 Holes
Drill after stud is assembled in position
Drill Thru-2 Holes

FIG. 323.—Sheldon worm and worm wheel carrier (5- to 6-ton. Dimension encircled for 1½-ton).

axles, also the Hotchkiss drive. The housing is a one-piece steel casting, details of which are shown in Fig. 322. Section *D D* is taken through the center of the axle; here the metal thickness is ⅜ in., gradually increasing ¾ in. under the spring seat, where the section is square, as seen in sectional view *A-A*. This axle is designed for use with springs having a width of 4 in., and spring centers of from 43 in. minimum to a maximum of 46 in. The advantage of a one-piece axle is that it forms a permanently

rigid construction. The section is gradually increased in thickness in order to distribute the stresses set up in the metal more uniformly. Figure 323 is a detail of the worm and wheel carrier,

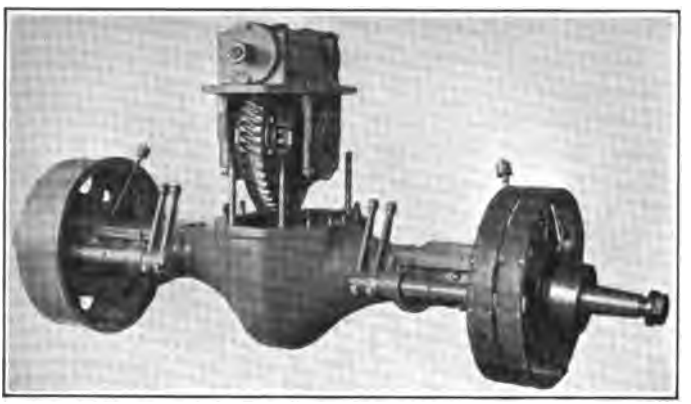


FIG. 324.—Sheldon rear axle.

which can be removed from the axle housing, as a unit, with the worm gear and the differential (see Fig. 324). The carrier is a malleable casting, the metal thickness being for the most part

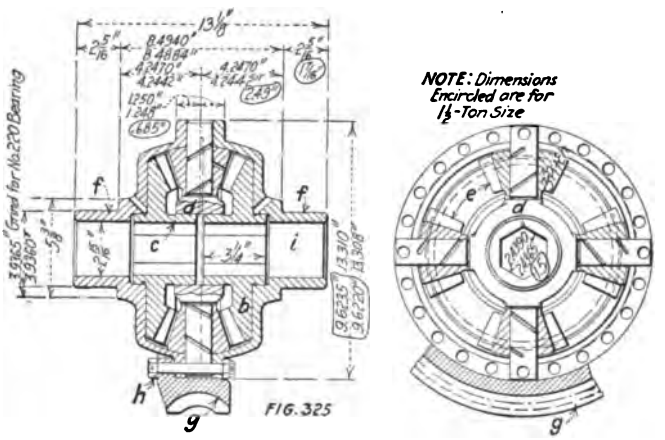


FIG. 325.—Differential assembly (Sheldon 5- to 6-ton).

⅜ in. except where strengthened, as for instance at the flanges. It is attached to the housing by 16 cap screws (12-⅝-in. and 4-⅜-in.). Figure 325 is the differential assembly; it consists of two bevel gears *b* having an extended hub *c* which is inserted

into and bears inside the bore of the spider *d*. Four bevel pinions *e* are seated on the spider arms and mesh with the two bevel side gears. The differential gears and pinions are made of high grade steel, heat treated. Their face width is 2 in. and their pitch 3. The differential spider, on which the pinions revolve, has $1\frac{3}{8}$-in. arms; it is a drop forging, carbonized and ground to size.

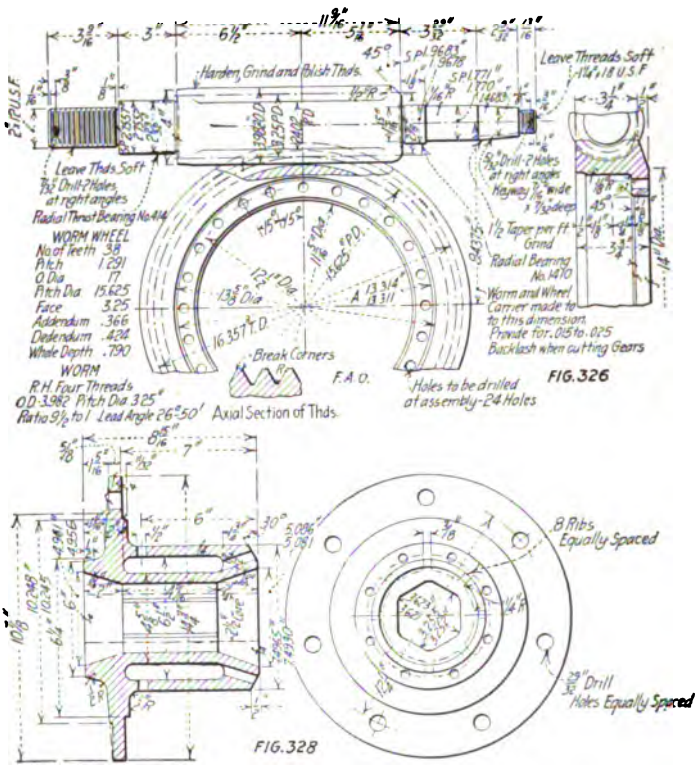


FIG. 326.—Worm and worm wheel. FIG. 328.—Rear wheel hub (Sheldon, 5- to 6-ton).

The two halves of the differential housing are bolted together and to the worm wheel *g* by means of 24-$\frac{9}{16}$-in. nickel-steel bolts. The extended hub *f* of the differential housing is rotating in double row ball bearings as seen from the assembly of the rear axle. The axle shaft extends through *i*, and is connected with the two bevel side gears by means of hexagonal ends.

Figure 326 is a detail of the worm and worm wheel. The

worm is of the straight or David Brown type, made of high grade steel, drop forged, hardened and polished, while the worm wheel is made of special bronze. The thickness of the flange is ⅞ in. and there is an extra thickness under the teeth. A detail of the worm driving end to which the universal joint is attached is shown at *a* in Fig. 321. As can be seen, the diameter of the shaft is 1¾ in. The lower section of the worm wheel runs in a bath of oil in the bowl of the axle housing. As the worm wheel rotates, all the bearings and the worm and worm wheel are lubricated by means of splash. The only other parts needing lubrication in the entire rear axle are the brake rocker shafts and the wheel bearings, and these are lubricated by oil cups.

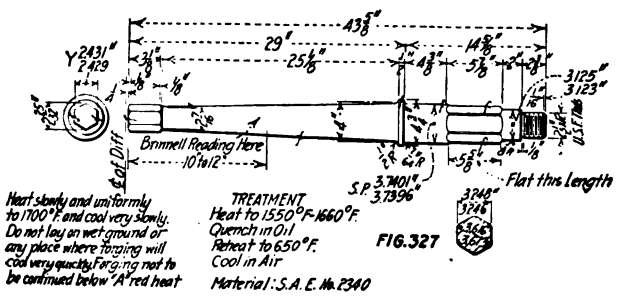

Brinnell Reading Here
10 to 12
Flat this length
Heat slowly and uniformly to 1700°F. and cool very slowly. Do not lay on wet ground or any place where forging will cool very quickly. Forging not to be continued below "A" red heat
TREATMENT
Heat to 1550°F-1660°F.
Quench in Oil
Reheat to 650°F.
Cool in Air
Material: S.A.E. No. 2340
FIG. 327

FIG. 327.—Drive shaft (Sheldon 5- to 6-ton rear axle).

Figure 327 is a detail of the drive shaft or axle made of S.A.E. 2340 steel, heat-treated. The shafts are forged to size from small billets. The driving ends of the axle are hexagonal. All the dimensions are given. The method of attachment to the wheel can be seen from the assembly drawing. The wheel is seated by collets *c* which are jammed by tightening the axle nut, and this is locked by the hub cap. The latter is attached to the rear hub flange by three ½-in. studs and lock washers.

Figure 328 is a detail of the rear wheel hub, which is a steel casting. The number of holes for the bolts in the flange vary from 6 to 8, depending on the number of spokes in the wheels, one bolt being provided at every second spoke. Figure 329 is the rear hub flange, and Fig. 330 the rear wheel hub cap, which is also used as a wheel puller. Figure 331 is the forged spring pad, which fits over the square part of the axle housing. Two of these are used for each spring, one under the spring direct, and one under-

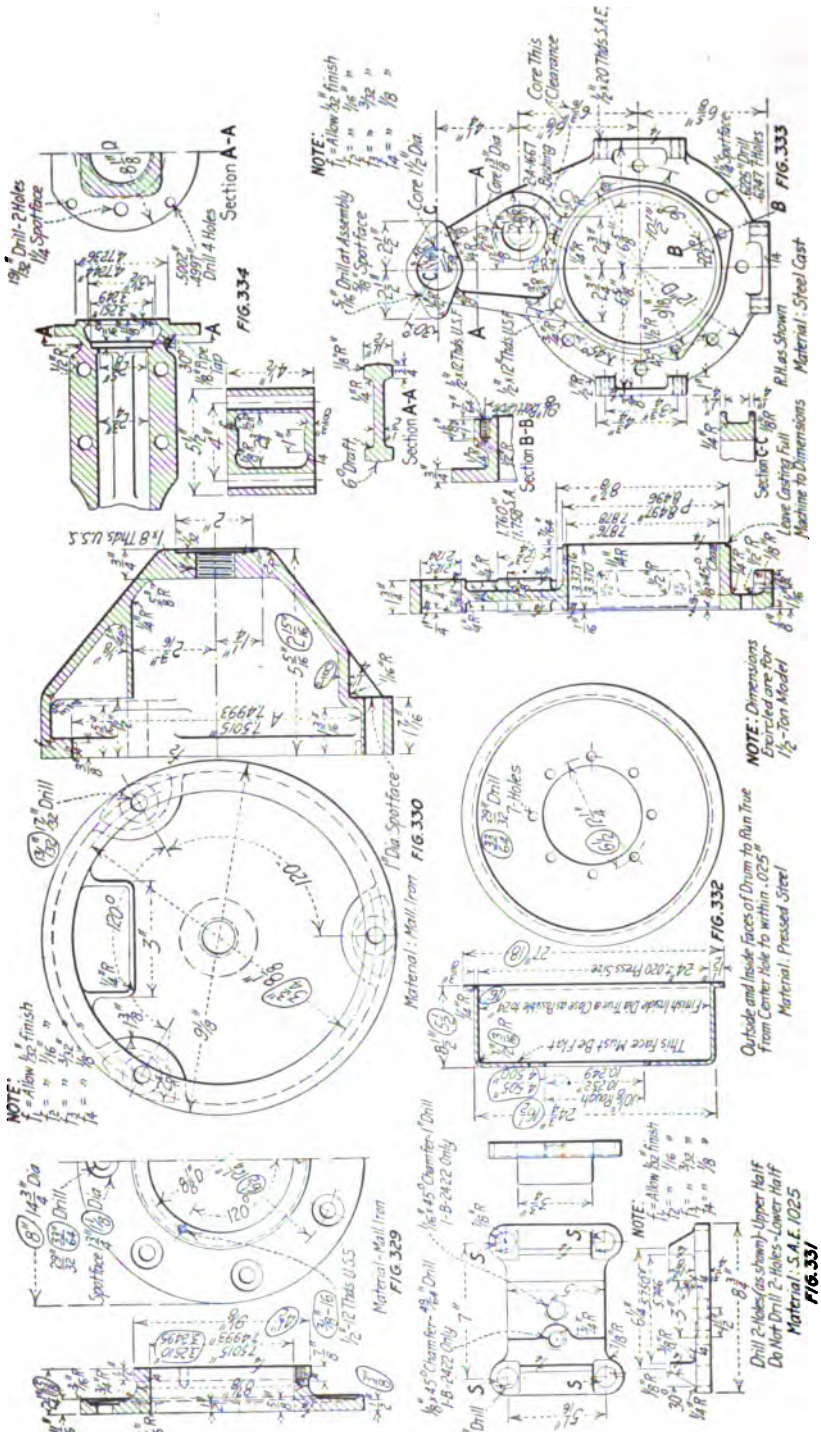


FIG. 329.—Rear hub flange. FIG. 330.—Hub cap. FIG. 331.—Spring pad. FIG. 332.—Brake drum. FIG. 333.—Brake spider (right hand) (Sheldon 5- to 6-ton axle details). FIG. 334.—Integral spring pad for 1½-ton axle.

neath the axle housing. Figure 332 is a detail of the Brake drum, a steel stamping, the metal thickness being ⅜ in.; the width of the drum is 8½ in. and its diameter 24¾ in. The two brakes are placed side by side in the drum. In the Sheldon axles the cam type of brake is used in the smaller sizes, while in the larger sizes the wrap-up type is employed. The brakes are so designed that there is no adjustment in the brakes themselves but in the parts which pull the brake levers, thereby simplifying the brake adjustment. In some types the adjustment is made by a sector attached to the brake pull lever. To tighten the brakes it is necessary only to take out the bolt which fastens the lever to the sector and move it to another hole of the sector.

The actual brakes are 3 in. wide each, and 24 in. in diameter. They are of the internal toggle wrap-up type. The brake levers are extended inside the frame. The brake levers, shafts and sleeves are serrated to allow for easy adjustment. A mushroom adjustment (see *Z*, Fig. 321) is provided on each brake band support bracket that the proper clearance may be obtained between the linings and the drum.

Figure 333 is a detail of the right hand brake spider which is a steel casting. The thickness of the metal can be seen from the section drawings.

Ratio.—The standard ratio in this rear axle is 8.75 to 1, but 10.25 to 1 can be supplied by the manufacturers when desired.

The following is a table of gear ratios, hp. allowable in the motor, and speeds, as specified by the manufacturer.

Gear ratio	Maximum hp. allowable	Speed in miles per hour at 1,000 r.p.m.	
		36-in. wheels	40-in. wheels
8.75 to 1	58	12.3	13.5
10.25 to 1	50	10.5	11.6

The above figures apply when this axle is used under a conventional motor truck of a 5-ton carrying capacity, with a standard transmission, having a reduction of 4.84 to 1 in the low gear. Hp. is figured by the Dendy-Marshall formula, as given below. Note in the table of hp. allowable with

different ratios, that, regardless of the ratio used in the axles, the figure obtained by multiplying the worm-gear ratio, hp. rating, and the transmission low gear reduction (4.84 to 1) does not exceed 2,450. Therefore, if it is desired to increase the low gear reduction in the transmission, either the motor size or the worm-gear ratio must be reduced so that product of the worm-gear ratio, rated hp. and transmission low-gear reduction does not exceed 2,450.

Let C = 2,450.
T = Low gear ratio (first speed) in transmission.
A = Gear ratio in axle.
HP = Horsepower of motor (at 1,000 r.p.m.) by Dendy-Marshall formula.

Maximum hp. allowable (at 1,000 r.p.m.) $= \frac{C}{T \times A}$.

Lowest gear ratio allowable in axle $= \frac{C}{T \times hp.}$.

Maximum low gear ratio (first speed) allowable in transmission $= \frac{C}{A \times hp.}$.

The Dendy-Marshall formula is $hp. = \frac{D^2 \times L \times N}{12}$, where D *is the bore, L the stroke, and N the number of cylinders, the speed of the motor being taken at 1,000 r.p.m., and it is claimed that these figures represent the average result obtained from American engines designed for truck motors.*

The manufacturers do not approve their equipment for higher powered motors or for lower reductions, except as stated, (if one is increased the other must be reduced), because the parts are designed for a given stress, and by increasing the hp. or the ratio, the torque is increased and this induces a higher stress in the parts than that for which they are designed.

The dimensions encircled in Figs. 321, 322, 323, 325, 329, 330, and 332, show those corresponding to the parts of the 1½-ton truck axle, made by the Sheldon Co. The dimensions of this axle and the bearings are as follows: Wheel bearings, #311; worm shaft front bearing #1407; rear bearing #403; differential bearings #215. Overall from end to end of hub caps 68 in.;

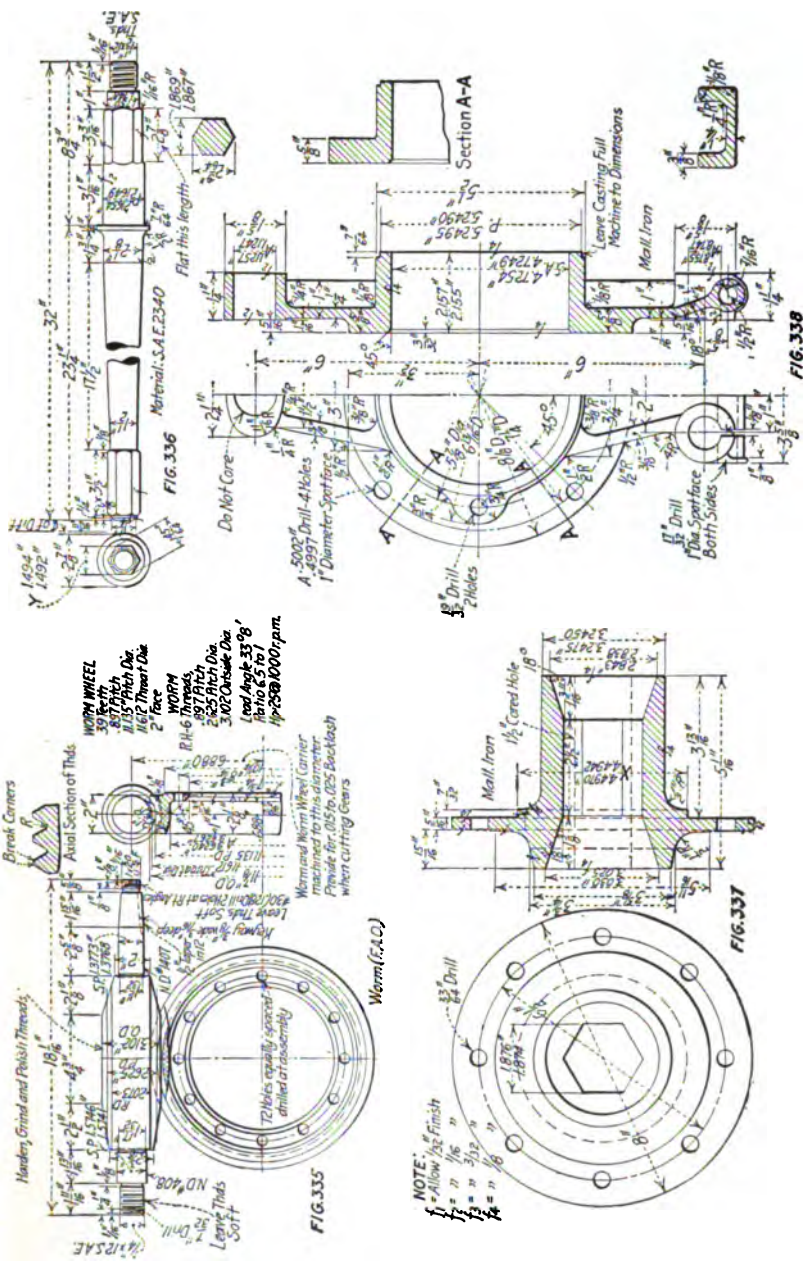


FIG. 335.—Worm and worm wheel. FIG. 336.—Drive shaft. FIG. 337.—Rear whee hub. FIG. 338.—Brake spider (Sheldon 1½-ton rear axle).

center line of axle to top of worm shaft housing 9 3/16 in.; center line of axle to bottom of axle housing 7 in.; horizontal distance from center line of axle to rear end of worm shaft housing 9 in.; outside diameter of hub flange 8 in.; diameter of spoke bolts ½ in.

The maximum load allowable on the rear tires is 4,000 lb., which includes weight of truck and pay load. The weight of the axle complete is approximately 550 lb. Figure 334 shows that instead of separate spring pads they are made integral; the thickness of metal changes from ¼ in. at the center of the housing to ½ in. under the spring seats. The differential spider has

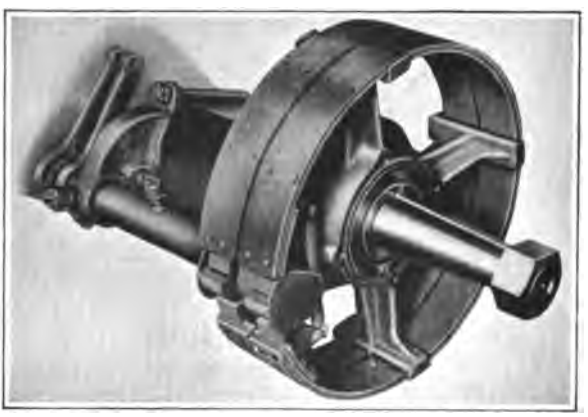


FIG. 339.—Sheldon cam type rear axle brakes, for 1½-ton trucks.

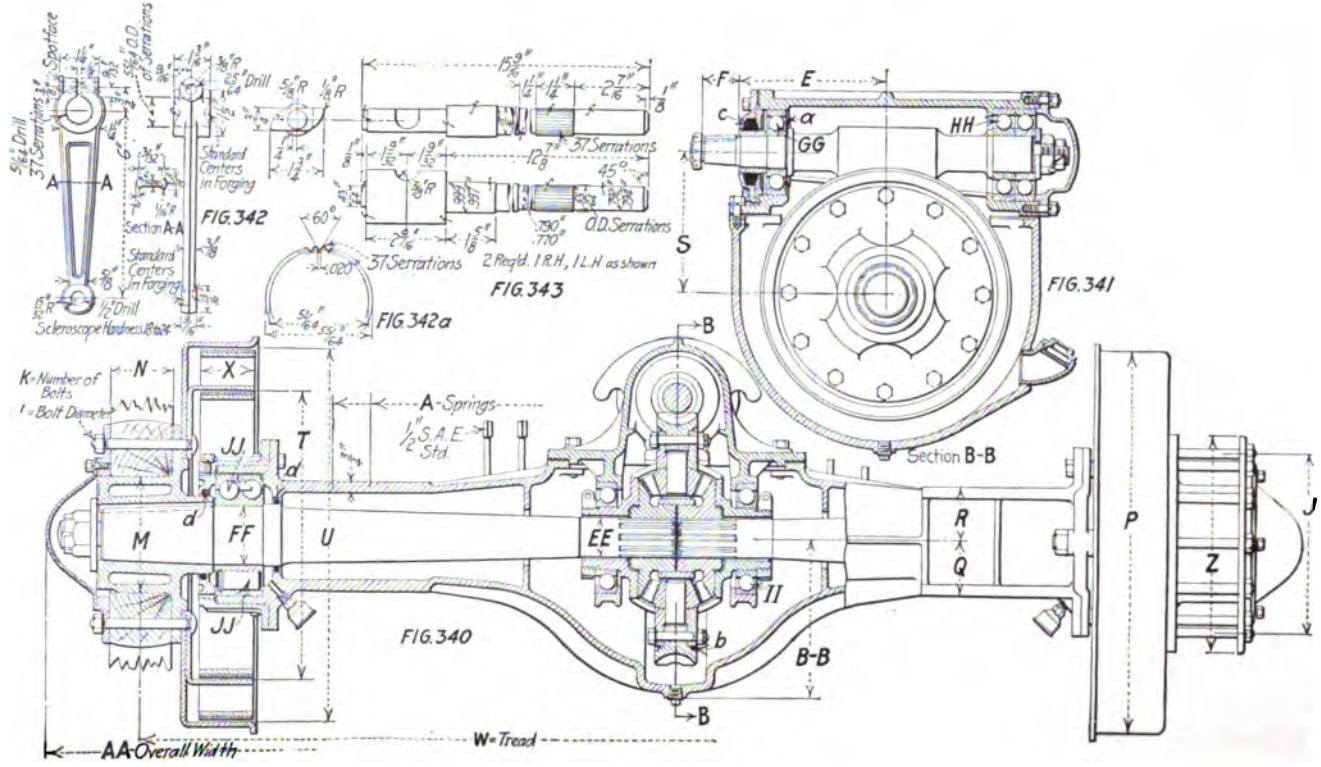
1-in. arms, while in a 5-ton axle they are 1¼ in. in diameter. Figure 335 gives the details of the worm and worm wheel, the distance between centers being 6.880 in. The worm wheel is attached to the differential housing by 12, ½-in., bolts. Figure 336 is the axle shaft, made of 2340 nickel steel, having a diameter at the bearing seat of 2 11/64 in. and 1½ in. across the flats of the hexagonal end at differential. Figure 337 is the rear hub, having a flange thickness of 5/16 in. The thickness of the metal of the brake drum (see Fig. 332) is ¼ in., each of the two brakes being 2 in. wide and 16 in. in diameter. The drums are turned on the inside to insure symmetry. The brakes in this model are of the cam type, as shown in Fig. 339. Figure 338 is a detail of the brake spider, the metal thickness being here ¼ in., except at the flanges and ribs.

The following is a table of the gear ratios, hp. allowable in the motor, and the speeds:

Gear ratio	Maximum allowable hp.	Speed in miles per hour at 1000 r.p.m.	
		34-in. wheels	36-in. wheels
6.5 to 1	25	15.7	16.6
7.8 to 1	21	13.0	13.8

In this axle the constant C (obtained by multiplying the low gear ratio of the transmission, the axle ratio and the hp.) is not to exceed 600. (See page 264 for full explanation.)

Figures 340 and 341 show the worm drive rear axle, semi-floating, manufactured by the Wisconsin Parts Co. The worm and worm gear complete are mounted to the differential carrier, inserted into the housing from above and attached thereto. The worm shaft is mounted on ball bearings, the forward one being free to float to allow for any elongation of the worm shaft due to a rise in temperature; the outer race of the bearing is therefore not confined endwise. The wheel is mounted on double row ball bearings or on straight roller bearings. The worm gear is held between the differential flanges and piloted (shown at *b*), thus supported throughout its circumference. At *a* is a steel disk, an oil spinner, clamped between the front worm bearing and the shoulder of the worm, which, as it spins around, throws the oil off, hence permitting only very little oil to pass through the ball bearing; the packing gland at *c* prevents leakage or egress of this oil. Velumoid gaskets are used at all the joints. To prevent egress of oil through the wheel bearing and upon the brakes, two felt washers *d* are provided, one just behind the wheel bearing and one in the adjusting nut. The brakes, as may be noted, are both internal, mounted on a *double* brake drum, the drums placed concentrically, thereby permitting the use of comparatively narrow drums. This enables the wheels to be placed closer to the load, *i.e.*, to the spring seats. The brake shafts run in bronze bushed bearings, which are provided with oil or grease cups, or they may be equipped with oilless bushings. The various dimensions of the different capacity trucks manufactured by this company are shown in the following table:



Figs. 340 and 341.—Wisconsin worm drive rear axle. Fig. 342.—Brake lever. Fig. 342a.—Brake cam. Fig. 343.—Service brake shaft (1- to 3½-ton).

WISCONSIN REAR AXLE DIMENSIONS

	Models						
	800G	800H	800J	900C	900D	900E	1,000B
Capacity, tons	1	1 1/2	2	2 1/2	3 1/2	3 1/2	5
Maximum spring load	4,000	5,500	7,300	8,800	12,000	12,000	16,000
A	38–40	38–40	38–40	38 1/2–41	39–40	43–44 1/2	40 3/4–46
D	7 1/4	7 1/4	7 1/4	8	8	8	9 1/2
E	6 15/16	6 15/16	7 13/32	8 9/16	8 9/32	8 9/32	9 23/32
F	2	2	2	2	2 1/4	2 1/4	2 1/4
G	1 3/8	1 3/8	1 1/2	1 1/2	1 3/4	1 3/4	1 3/4
H	11 5/8	11 5/8	11 5/8	12 1/8	12 1/8	14 1/8	10 1 1/16
J	7 5/8	7 5/8	8	9	10	10	11 1/2
K	14	14	14	14	14	14	12
L	7/16	7/16	7 1/16	1/2	9/16	9/16	5/8
M	3 5/8	3 5/8	4	4	4 1/2	4 1/2	7 1/8
N	2	2	2 1/4	2 1/2	3 1/2	3 1/2	4
O	2 5/32	2 5/32	2 5/32	2 9/32	2 9/32	2 9/32	1 1/32
P	17 3/8	17 3/8	17 3/8	20 1/2	20 1/2	20 1/2	24 5/8
Q	2 1/2	2 1/2	2 1/2	2 3/4	3	3	3 1/2
R	2 1/2	2 1/2	2 1/2	2 3/4	3	3	3 1/2
S	6	6 5/8	7 1/2	7 1/2	7 3/4	7 3/4	9 1/16
T	12 1/4	12 1/4	12 1/4	15 5/16	15 5/16	15 5/16	18 1/2
U	16 7/8	16 7/8	16 7/8	20	20	20	24
W	57 1/2	57 1/2	58	59 1/2	60	64	70 1/2
X	2 1/2	2 1/2	2 1/2	2 1/2	2 1/2	2 1/2	3 1/2
Y	3	3	3	3	3 1/2	3 1/2	4
Z	9	9	10	10 1/2	12	12	14
AA	66 1/4	66 1/4	67	70	71 1/2	75 1/2	82 5/8
BB	6 7/8	7 3/4	8	8 1/4	8 1/4	8 1/4	10 1/4
Standard reduction	7 3/4–1	8 1/4–1	8 3/4–1	8 3/4–1	10–1	10–1	11 3/4–1
Optional reduction	62–1	7 1/4–1	7 3/4–1	9 3/4–1	9–1	9–1	
	8 3/4–1	9 1/4–1	9 3/4–1	10 3/4–1	10 3/4–1	11 3/4–1	
Maximum torque on low gear, lb.-feet	400	400	500	700	800	800	1,000
Weight, pounds	500	640	700	875	975	1,000	1,590
EE, inches	1 9/16	1 3/4	1 7/8	2	2 1/4	2 1/4	2 1/2
FF, inches	2 3/16	2 3/8	2 1/8	3	3 3/8	3 3/8	3 3/4
GG	407	407	309	310	311	311	411
HH	407[1]	407	408	410	411	411	412
II	212	213	213	214	215	215	220
JJ	311	311	313	315	317	317	319

[1] Single bearing at rear also in 800G model.

The housings and other cast parts of these axles are malleable castings. During inspection, the housings are mounted on the spring seats, under a press, and a pressure of from 10- to 25-tons (depending on the size) is applied at the center, to test for, and eliminate, imperfect castings. The drive shafts are made of chrome-nickel steel, S.A.E. 3140, properly heat treated, to show

an ultimate strength of about 200,000 lb. per square inch. At the wheel end they have a taper of ¾ in. per foot, while at the other end they are provided with S.A.E. standard splines. The worms are made of alloy steel, high in chromium and manganese; after annealing, the forgings are machined, then carbonized 1/16 in. deep and heat treated, thereby obtaining a very tough core and an extremely hard wearing surface. The final operation is the grinding of the tooth flanks and the bearing seats. The worm gears are cast in metal molds from a titanium alloy.

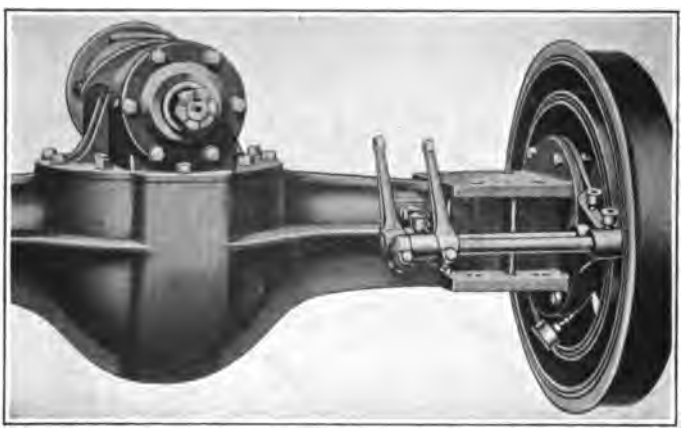


FIG. 344.—Wisconsin rear axle.

Figure 344 is an illustration of the rear axle; the brake levers and the concentric brake drums are in full view. The outer drum is for the foot-operated service brake, while the inner drum is for the emergency brake. Figures 342 and 343 are details of the brake lever and the service brake shaft respectively, of the 1-ton to 3½-ton models. In the 5-ton model, the length of the lever is 8 in. instead of 6 in., and the outside diameter at the serrations is 1 7/64 in.

Brake adjustments are made by serrated levers and shafts; the bolt in the lever hub is loosened, placed in the desired position, and then the bolt is again tightened. The brakes are cam operated, the cams being made integral with the brake shafts, and the cam outline may be perceived on Fig. 342*a*.

Figure 345 shows the axle constructed by the Timken-Detroit Axle Co. in their models of from 2- to 5-ton capacities. In these models this company uses the full-floating axles while in their

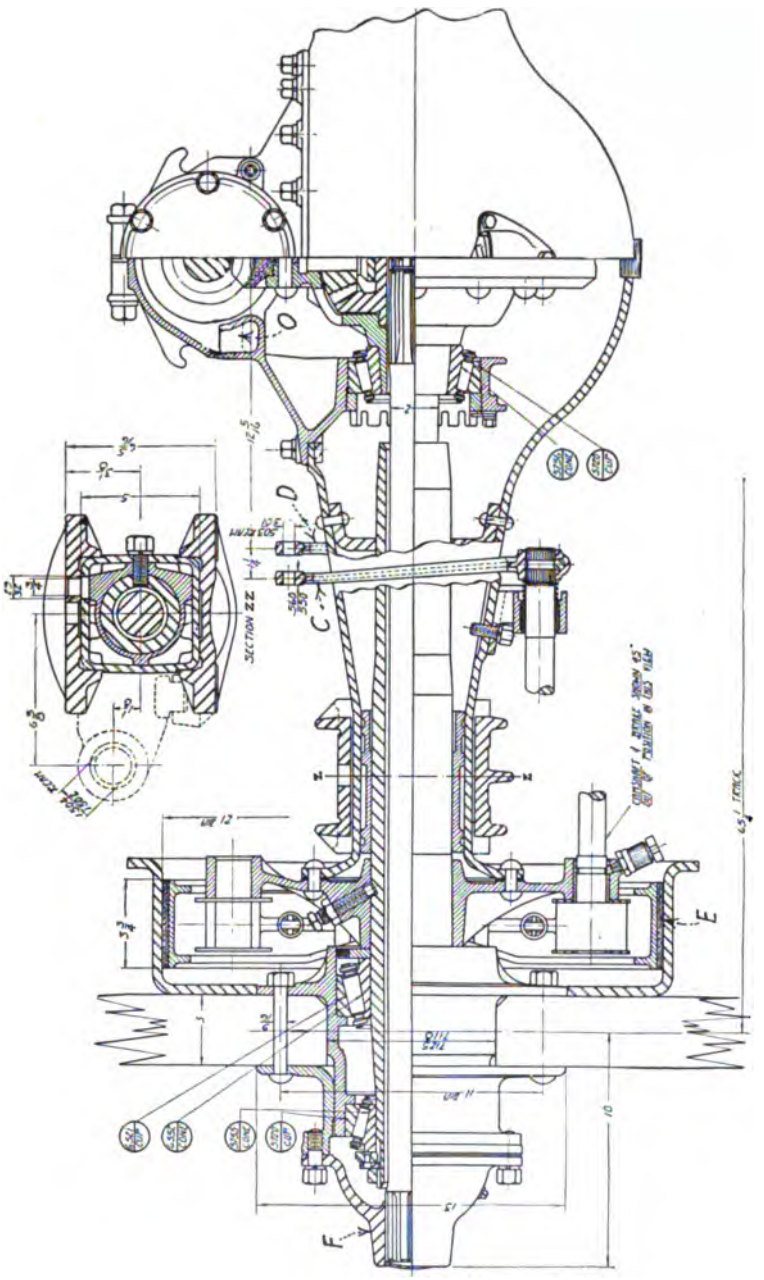


FIG. 345.—Timken-Detroit worm drive rear axle, full-floating, for 3½-ton truck.

smaller models, capacities of from ¾- to 1½ ton, they use the semi-floating type. The dimensions given in this drawing refer to the commercial 3½-ton truck.

All the Timken axles are worm driven, and the entire construction is similar to the class B axle, the chief difference being in the brakes. Figure 346 is a side view of the brake shoes in the brake drum. There are four shoes, marked *p*, *q*, *r*, and *s*, each extending about one-fourth of the circumference of the drum,

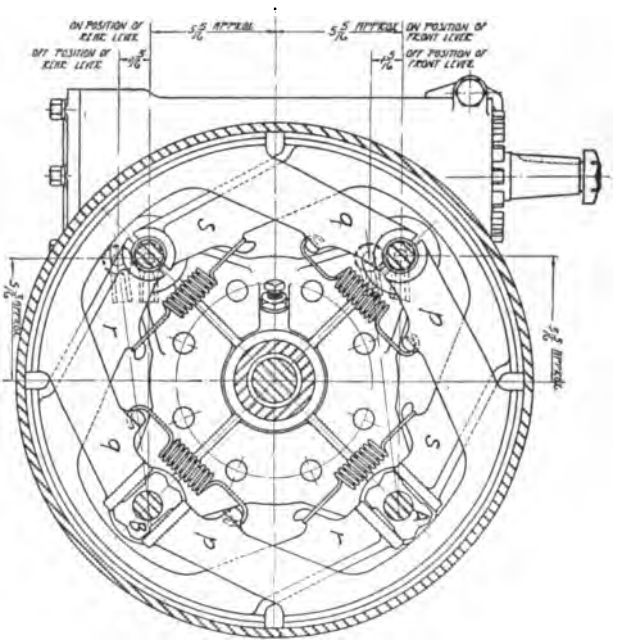


FIG. 346.—Timken-Detroit rear axle; brake shoes.

but to the full width, as seen from the assembly at *E*. The brake cams *A* and *B*, which are operated by the brake shaft levers *C* and *D* respectively, actuate two opposite shoes; *B* actuating *p* and *q*, and *A* operating *r* and *s*.

The brakes in the 3½-ton model are 21 in. in diameter and 3½ in. wide. The outer edge of the brake drum is flanged to prevent distortion under pressure.

Housing.—The housing is of one piece, of rectangular section at the ends. In the full-floating models, nickel steel, heat-treated sleeves are carried in, close to the differential, being

supported near the inner end by rigid reinforcing plates. At the outer ends, these sleeves carry Timken roller bearings for the wheels. The brake spiders are riveted to the ends of the housing.

Spring Seats.—Two-piece spring seats are supplied, either with or without bosses for radius rod connections.

Differential Carrier.—The oil feed gutters *O* at the sides of the worm, feed the lubricant to the back of the bearings; oil holes are provided with small drain plugs on either end which can be removed for cleaning these passages. The forward bearing cup (see class B rear axle drawing) is mounted in a cup carrier so that by adjusting the cup carrier the bearing cup moves as an integral part of it.

Differential.—The hole in each hub of the differential case accommodates the increased diameter of the inner end of axle shaft which is made with ten splines.

Shafts.—The shafts are forged from one piece of nickel-chrome steel, splined on both ends, heat-treated and ground to accurate size. They are slightly smaller in diameter at the center. The inner ends of the shafts enter the splined differential while the driving plates *F* on the outer ends are separate forgings splined and shrunk in place.

Worm and Gear.—The Timken-Detroit worm is of nickel-chrome steel, heat-treated, hardened, and accurately ground. The gear wheel is of a special bronze.

Lubrication.—As in all Timken-Detroit worm drive axles, oil is carried in the central housing at the level of the filling thimble. When the worm wheel is in motion it carries oil over the worm whence it is caught in gutters *O* on the inside of the carrier. It then flows by gravity to the back of the bearings, through them, and back to the housing.

The worm gears and the worm bearings are the same as that shown in the class *B* rear axle. The adjustments for the worm bearings, as recommended by the manufacturers, are as follows: "To adjust the worm shaft bearings remove the packing gland from the carrier to avoid binding on the shaft and to allow free movement of the shaft. Next remove the lock from the slot and with a flat tool, screw in the forward bearing adjusting ring, until the shaft binds. Now unscrew the ring again several notches to allow end play so that the shaft can expand when it becomes heated under running conditions. The exact

amount the ring should be backed off is shown in the following table:"

Size Axle, Tons	Play in Inches	Number of Notches
¾ to 1½	.011 to .015	3
2 to 3½	.016 to .020	4
5	.020	4

The following are the rules for the differential bearings (see Fig. 347):

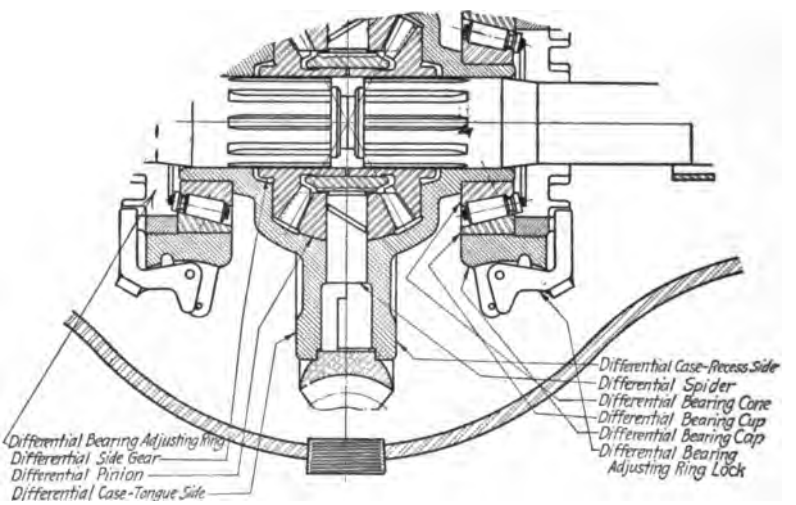


Fig. 347.—Timken-Detroit differential.

"In adjusting the differential bearings a combination adjustment is necessary; first, all play must be taken out of the bearings by screwing up the adjusting rings on both sides of the differential, and second, the worm and worm wheel must be set in proper alignment by adjusting the differential bearings on both sides end for end, moving the differential and worm wheel to one side or the other as required for correct alignment.

"All ordinary adjustment of these bearings should be made on the left hand bearing (standing at the rear and facing toward the front of the truck); never disturb the right hand bearing except when it is necessary to take down the entire differential assembly. By making the adjustments on the left bearing only the alignment of the worm and wheel will not be destroyed. To adjust the bearings the differential carrier must be removed from the

housing; take out both shafts, remove the cap screws holding the carrier to housing and, having disconnected the propeller shaft, lift out the entire carrier assembly. In most cases it will be necessary to remove the axle from the chassis before the carrier can be pulled out.

Now release the lock on the left hand bearing adjusting ring and tighten up the bearings by screwing in this ring with a flat tool until all end play or shake in the differential housing has been removed. Having obtained a good tight adjustment unscrew one notch and lock in this position with the locking finger. If the

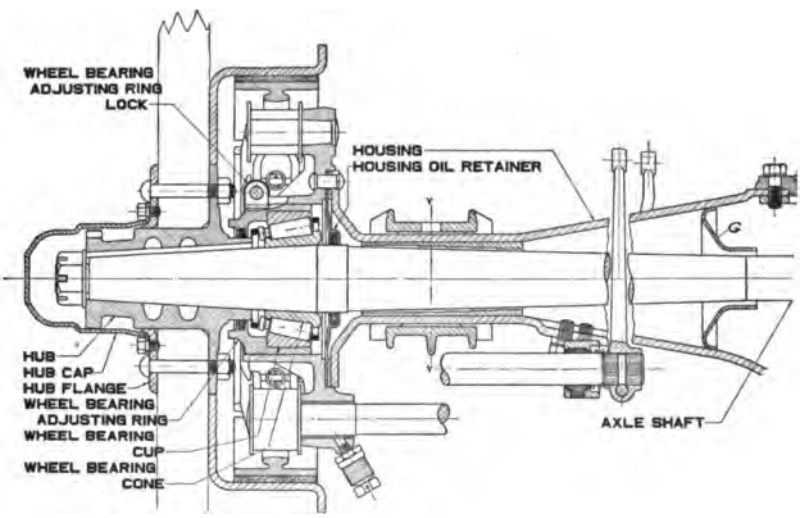


FIG. 348.—Timken-Detroit semi-floating rear axle, for ¾- to 1½-ton trucks.

alignment of the worm and wheel is not correct, unscrew one ring, at the same time tightening up on the other the same number of notches until the center of the worm wheel is directly under the center of the worm. Check up again for end play to be sure that the bearing adjustment is all right. The carrier assembly may now be replaced, the drive shafts put in, and the axle reassembled under the truck. In replacing the carrier assembly, be sure that you tighten up well on the cap screws holding the carrier to the housing.

Figure 348 is the fixed hub or semi-floating type of Timken rear axle. This differs from the former type in that the axle shaft is fixedly attached to the wheel direct, instead of by a

splined driving plate. Other changes are the oil guard *G* near the differential case, visible on the extreme right of the figure, and felt rings near the wheels, to prevent oil from reaching

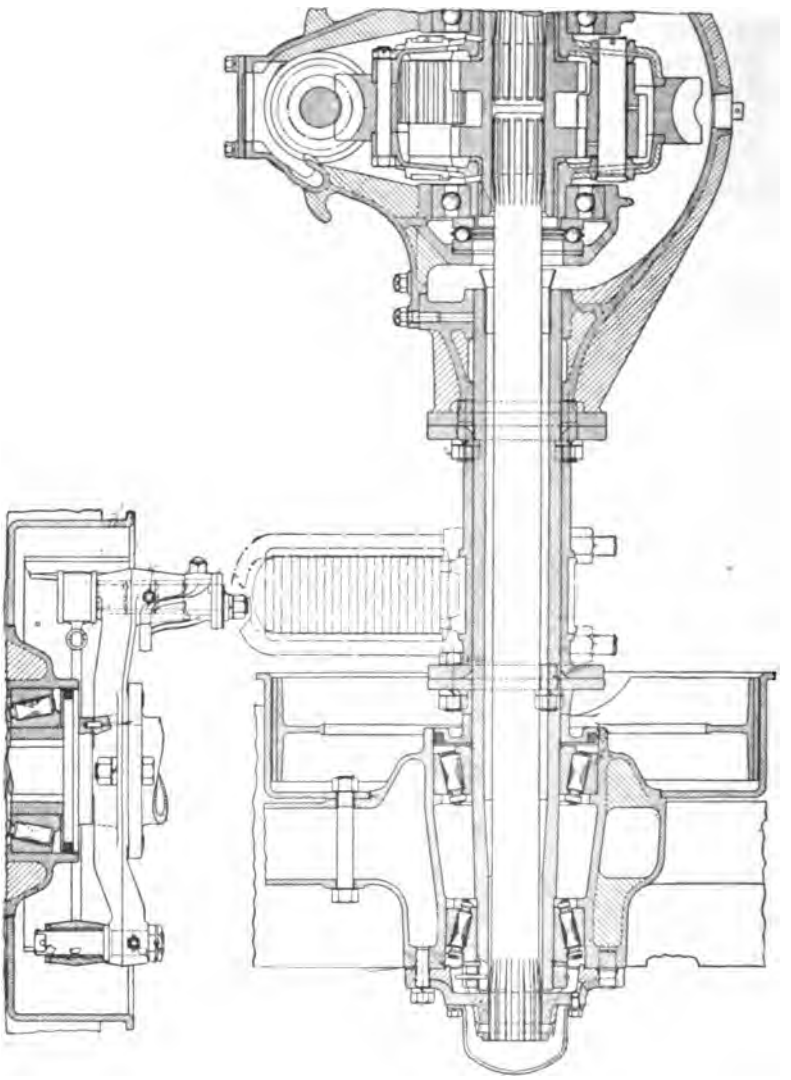


Fig. 349.—Pierce-Arrow 3½- and 5-ton truck rear axle (full-floating).

the brake drum. The housing of the fixed hub type is somewhat simpler and lighter since the axle tube, which extends to the outside of the wheels (see Fig. 345) is eliminated. In full-floating

axles there is a greater stress on the axle housing, near the wheel, than in the fixed hub type, as described on pages 199 and 335.

The Pierce-Arrow 3½-ton and 5-ton models are full-floating and are provided with taper roller bearings for the wheels, while the worm and worm wheel are supported on ball bearings. A propeller shaft brake is used as a service brake, while the emergency brake acts on the inside of brake drums attached to the wheels, as seen in (Fig. 349), which is a cross sectional view of the rear axle assembly. In their 2-ton model this firm makes use of taper roller bearings throughout (also for the worm), the axle is

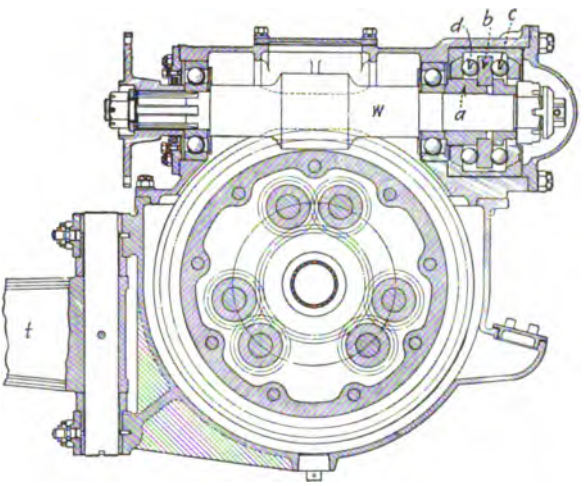


FIG. 350.—Pierce-Arrow worm mounting (for 3½- and 5-ton models).

semi-floating, and the two brakes are placed side by side in the rear wheel brake drums. Instead of driving through torque and radius rods, the Hotchkiss drive is employed in this small model.

Figure 350 shows the worm mounting of their larger models; the radial load is carried by single row ball bearings, one in the front and one in the rear, while the thrust in both directions is taken by a double row ball thrust bearing located at the back. Under ordinary forward running of the truck, the worm tends to move to the right, when the collar *a* exerts a pressure against ring *b* and thereby against balls *c*. When going backward, the balls *d* take the thrust. The differential bearings consist likewise of single row balls for supporting the radial load, and single row ball thrust bearings for the thrust load, as seen from Fig. 349.

Differential.—The differential consists of straight spur pinions and gears. There are three pairs of pinions of double width; the two pinions of each pair are not only in mesh with each other

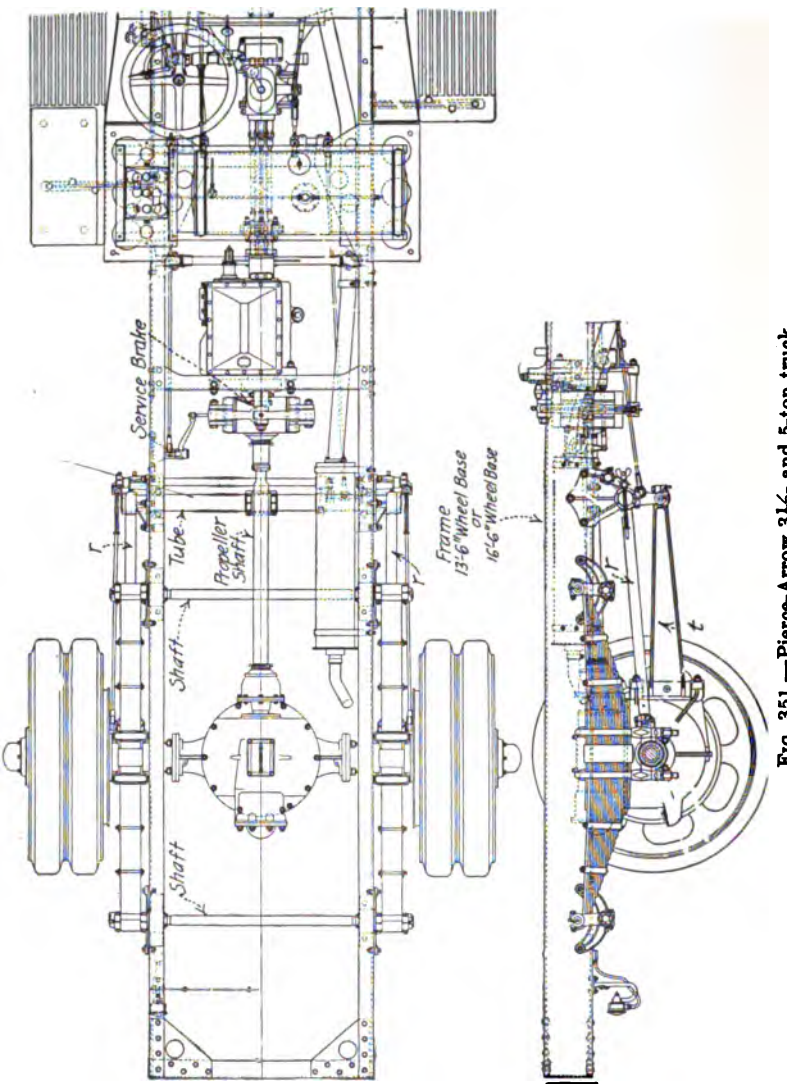


FIG. 351.—Pierce-Arrow 3½- and 5-ton truck.

but, in addition, one is in mesh with one side gear and the other pinion with the other side gear; the last two figures will make their construction plain.

In their larger models the Pierce-Arrow Co. uses a torque arm (marked *t* in Fig. 350) in the center of the axle, underneath the propeller shaft, to take the torque, and two radius rods which also serve the purpose of thrust rods, *i.e.*, they keep the axle at the same distance from the front anchorages of these rods when the axle moves up or down, and they thrust or push the vehicle forward through these rods. Figure 351 is a plan view and side view of a portion of the chassis; the torque rod *t* and the radius rods *r* can be seen. The propeller shaft brake is situated behind the transmission case which is suspended from a cross member in the back, and supported by one in the front.

Examples of Internal Gear Drives.—Figure 352 is an assembly drawing of the ¾-ton Torbensen axle. In this axle the load carrying member is of I-beam section; such a section can be made lighter for a given strength and a given direction of stresses than round or square sections. The objection sometimes cited against I-beam rear axles is that stresses are imparted to it from other directions than purely vertical; when striking an obstruction on the road, or when applying the brakes and locking the wheels, the stresses are not entirely vertical (the I-beam can best resist vertical stresses); however, by far the greatest stresses are substantially the same as in front axles, which are almost universally made of I-beam section. A detail of the I-beam load carrying member is given in Fig. 354. It is made of 1040 S.A.E. steel heat-treated, having a minimum tensile strength of 85,000 lb. and a minimum elastic limit of 60,000 lb. At the ends of the I-beam are tapered holes, 4¼ in. deep, shown at *x*, for inserted spindles (see Fig. 355) made of chrome-vanadium steel (S.A.E. 6130). The advantages claimed for such spindles is that they can be made to the exact hardness desired more easily, and by employing a higher grade alloy steel, heat-treated, the spindle diameter may be made smaller, which permits the use of smaller bearings. The spindle is ground to a tolerance of .0005 in. Another special feature of the Torbensen axle is that the jackshaft is carried behind and supported in the center of, the I-beam axle. At *A* in Fig. 352 can be seen how the differential housing of the jackshaft is shouldered where it is attached to the I-beam by means of cap screws (not shown). Locating the differential housing behind the load-carrying member permits the removal of the housing cover *B* for inspection and adjustment of the gears and bearings.

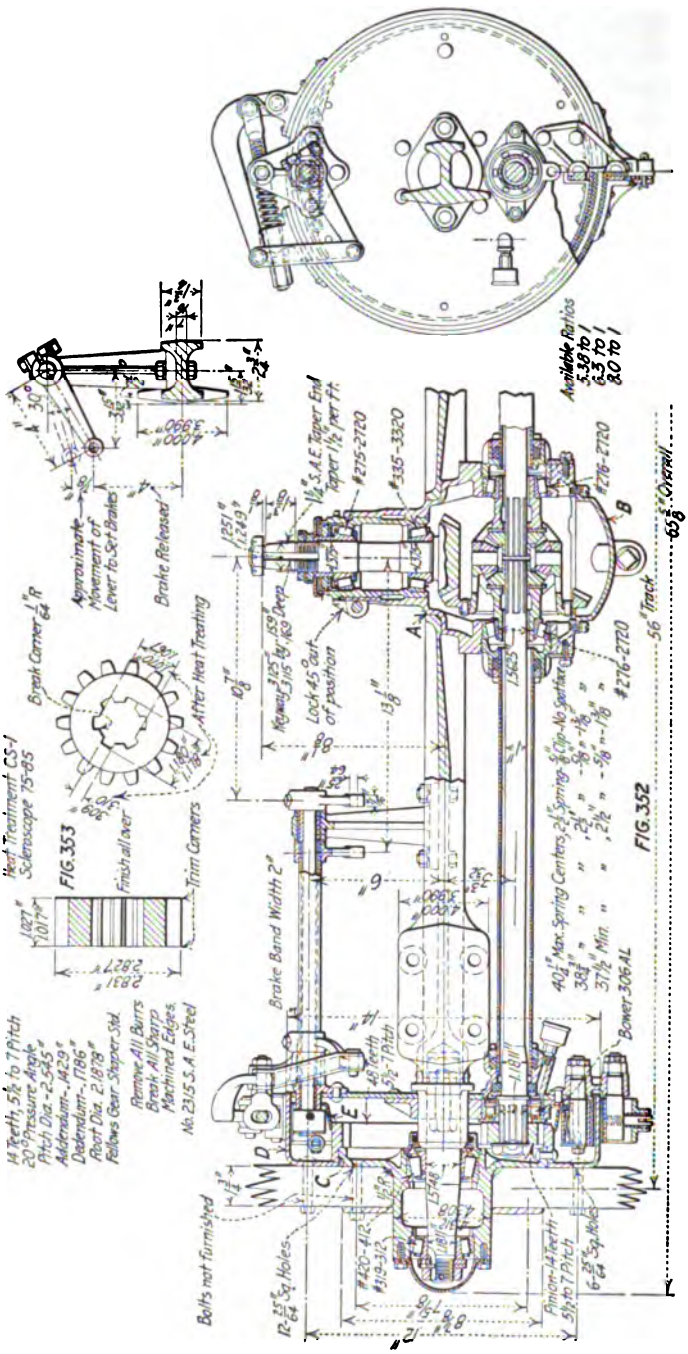


FIG. 352.—Torbensen ¾-ton internal gear drive rear axle. FIG. 353.—Spur gear driving pinion.

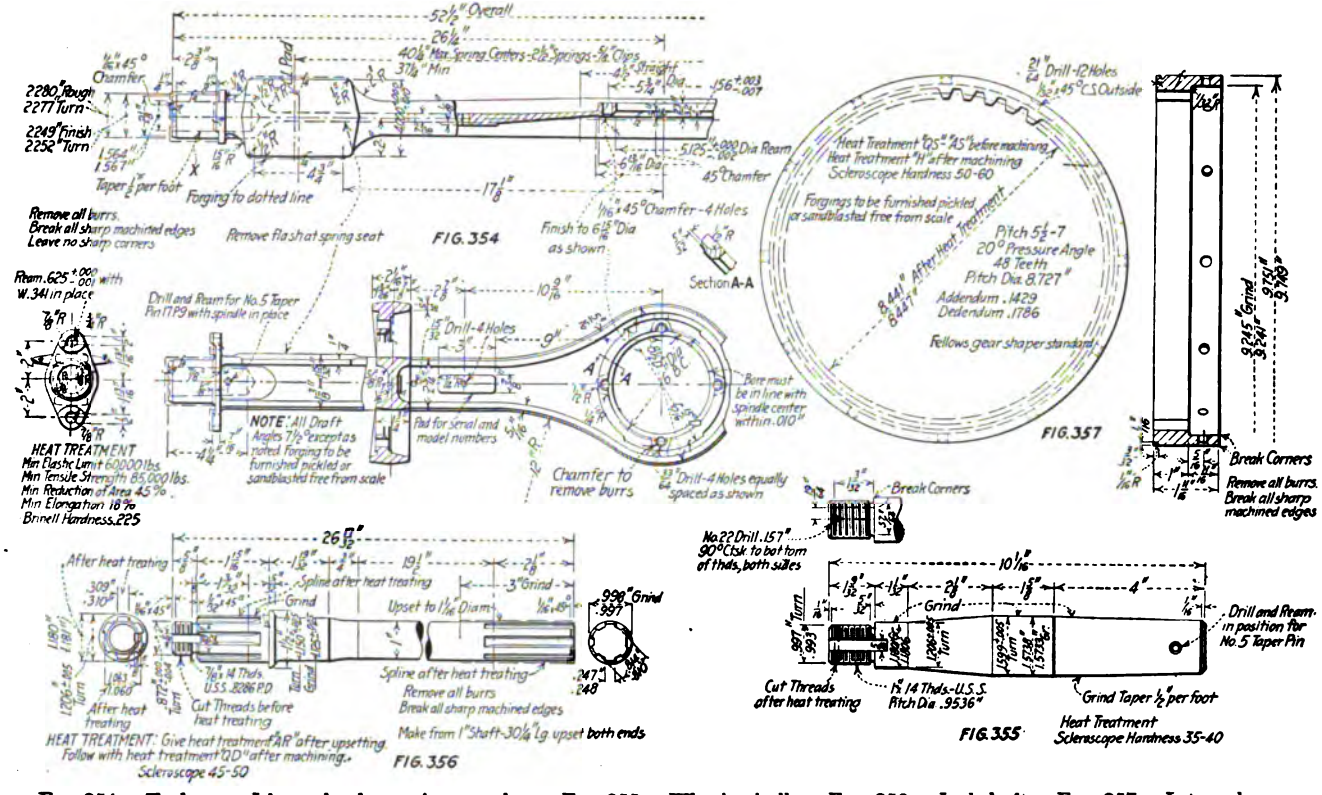

52½" Overall
26¼"
40⅛" Max. Spring Centers
37¼" Min
Forging to dotted line
Remove all burrs.
Break all sharp machined edges
Leave no sharp corners
Remove flash at spring seat
FIG. 354
Finish to 6 15/16" Dia as shown
Section A-A
Ream .625 +.001 -.001 with W.341 in place
HEAT TREATMENT
Min Elastic Limit 60000 lbs.
Min Tensile Strength 85000 lbs.
Min Reduction of Area 45 %
Min Elongation in 8"in 18 %
Brinell Hardness 225
Drill and Ream for No. 5 Taper Pin (TP) with spindle in place
NOTE: All Draft Angles 7½° except as noted. Forging to be furnished pickled or sandblasted free from scale
Pad for serial and model numbers
Chamfer to remove burrs
Bore must be in line with spindle center within .010"
Heat Treatment "QS" "AS" before machining
Heat Treatment "H" after machining
Sclerascope Hardness 50-60
Forgings to be furnished pickled or sandblasted free from scale
Pitch 5½-7
20° Pressure Angle
48 Teeth
Pitch Dia. 8.727"
Addendum .1429
Dedendum .1786
Fellows gear shaper standard
After Heat Treatment
FIG. 357
Break Corners
Remove all burrs.
Break all sharp machined edges
No. 22 Drill .157"
90° Ctsk. to bot. form of thds, both sides
Grind
Drill and Ream in position for No. 5 Taper Pin
Cut Threads after heat treating
1" 14 Thds-U.S.S. Pitch Dia .9536"
Grind Taper ½" per foot
Heat Treatment
Sclerascope Hardness 35-40
FIG. 355
After heat treating
Spline after heat treating
Upset to 1 1/16" Diam.
Cut Threads before heat treating
Remove all burrs
Break all sharp machined edges
Make from 1" Shaft-30¼" Lg. upset both ends
HEAT TREATMENT: Give heat treatment "AR" after upsetting.
Follow with heat treatment "D" after machining.
Sclerascope 45-50
FIG. 356

FIG. 354.—Torbensen I-beam load-carrying member. FIG. 355.—Wheel spindle. FIG. 356.—Jackshaft. FIG. 357.—Internal gear.

The advantage claimed for the feature of attaching the differential housing rigidly to the load-carrying member is that under deflection of the axle, the jackshaft will flex with it, and thus maintain the line contact of the spur gears, whereas if these members are not attached together rigidly, any strain in the load-carrying member will deflect it, and since the shaft with the pinions is not deflected, only the outer edges of the teeth will come in contact with the internal gears. The bevel gear drive pinion and the bevel drive gear are made of S.A.E. 2315 nickel steel, carbonized (heat-treatment G). They have a pitch of 5, and a face width of $\frac{7}{8}$ in. The drive gear is attached to the differential housing by 8, $\frac{5}{16}$-in., rivets. The pinion has 18 teeth and the gear 33, thus the ratio between the bevel gear is 1.83 to 1. Figure 356 is a detail of the chrome-nickel steel jackshaft (S.A.E. steel 3140) made from 1-in. round stock, both ends being "upset" to increase the diameter, as required. Figure 353 is the spur gear driving pinion, which is splined to the shaft and locked to it by a nut as shown in the assembly. It has 14 teeth, and is made of S.A.E. 2315 nickel steel, heat-treated. Figure 357 is a detail drawing of the internal gear made of 1050 S.A.E. steel (a comparatively high carbon steel). It has 48 teeth, hence the ratio of the internal gears is, $48/14 = 3.428$ to 1. The total ratio of the rear axle is the ratio between the bevel gears multiplied by the ratio of the internal gears, which equals $1.83 \times 3.428 = 6.273$ to 1. The internal gear is attached to the wheel hub by a press fit and 12, $\frac{5}{16}$-in., rivets, as shown at *C* in the assembly. This is claimed to be a distinct advantage over the constructions where the ring gear is held in place by bolts passing through the gear and the wood wheel, where the gear is seated on a wooden surface. Attaching the gear to the hub direct provides a metal to metal contact, which tends to keep the gears in line, while when it is attached to the wheels by bolts it is claimed that it is liable to distortion when drawing up the bolts in assembling. The brake drum *D* is mounted to the wheel to admit of some space between it and the internal gear, thereby allowing sufficient room for the brakes, which may both be placed inside the drum, if the latter is widened. In the drawing one internal and an external brake are shown, and a sheet metal ring *E* is employed to keep dust and dirt from the spur gears and to shield the brakes from the lubricant of the gears. The attachment of the brake shaft bracket to the I-beam axle is shown in the assembly.

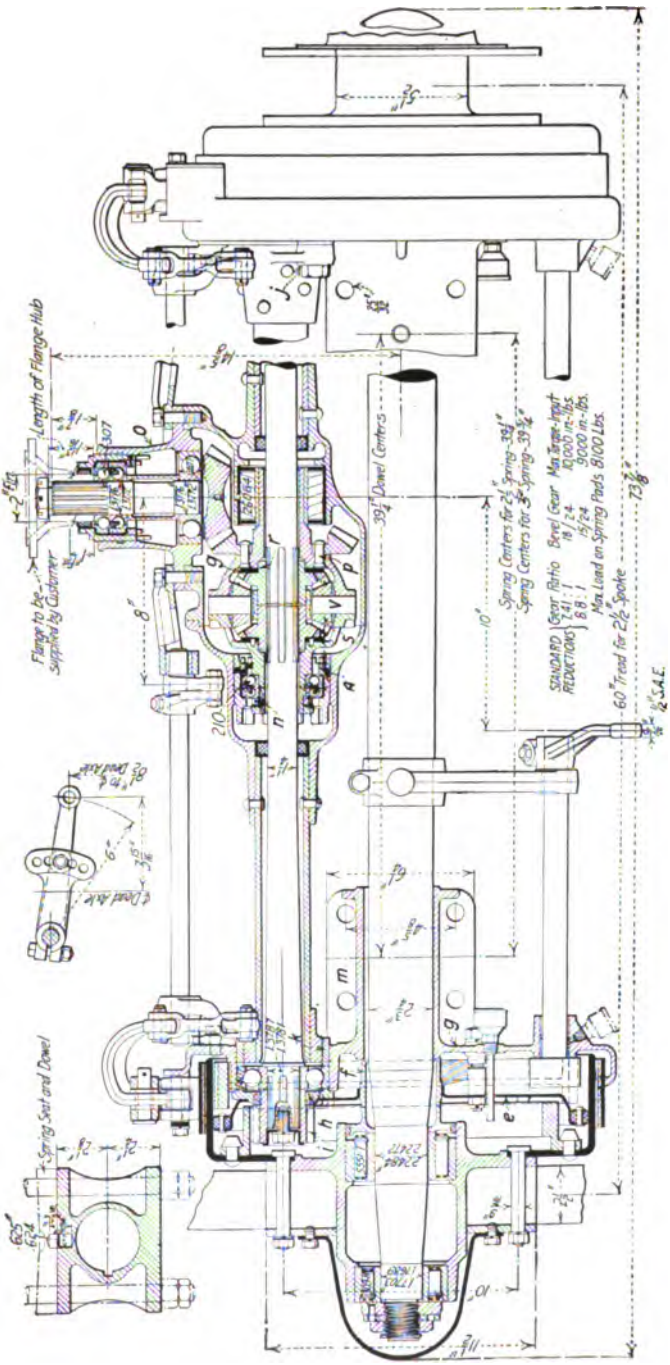


FIG. 358.—Russel 2-ton internal gear drive rear axle.

Weight.—It is stated by the makers that this ¾-ton axle is of about the same weight as a bevel-gear-drive axle suitable for a 3,000-lb. passenger car, due chiefly to the fact that the power transmitting parts are light, since the live axle and the bevel gears are not subjected to the maximum torque.

In the Russel internal-gear axle, (Figs. 358 and 359), a round axle, made from bar stock of chrome-nickel steel, is used. It is heat-treated at the mills, which is claimed to give identical physical properties throughout; this is difficult to obtain with forged axles which are subjected to local heating, and which are non-uniform in section. The load-carrying member is behind the jackshaft, which permits the use of a shorter propeller shaft, thus lessening weight and whip in this shaft. Furthermore, by having the jackshaft pinion in front of the wheel center, the pressure of the driving pinions on the internal gear is downward and this reduces the load on the wheel bearings considerably. If the pinions are behind the dead axle, the bearing pressures arising from the drive are added to those due to the static load on the wheel. (See page 137 showing effect of driving gear being on the right or the left of a driven gear.) It is claimed that under normal running conditions the load on the rear wheels is less than one-half of what it would be were the jackshaft placed behind the dead axle.

The jackshaft housing is entirely independent of the dead or load-carrying axle, and is not bolted to it. With all load carriers there is some deflection due to the strain in the metal, especially under severe service, and by not having the jackshaft rigidly attached to it, the strain of the load-carrying axle is not transmitted to the jackshaft. The jackshaft housing is rigidly attached at one end to the brake support, at *j*, and loosely on the other end, at *k*. The spring seat *m* which is integral with the brake support is keyed to the axle at the left side as shown; the other spring seat is free to turn. Thus a certain flexibility is obtained, when the load-carrying member is deflected under load or when the traction on one wheel should be greater than on the other, which, it is said, tends to prevent misalignment between the spur gear pinion and the internal gear. By having only a small reduction at the bevel gears, the diameter of the live axle or jackshaft can be made smaller than in other types, and yet possess a larger factor of safety. See pages 235 to 243 for a discussion of the advantages and disadvantages of the various types of axles.

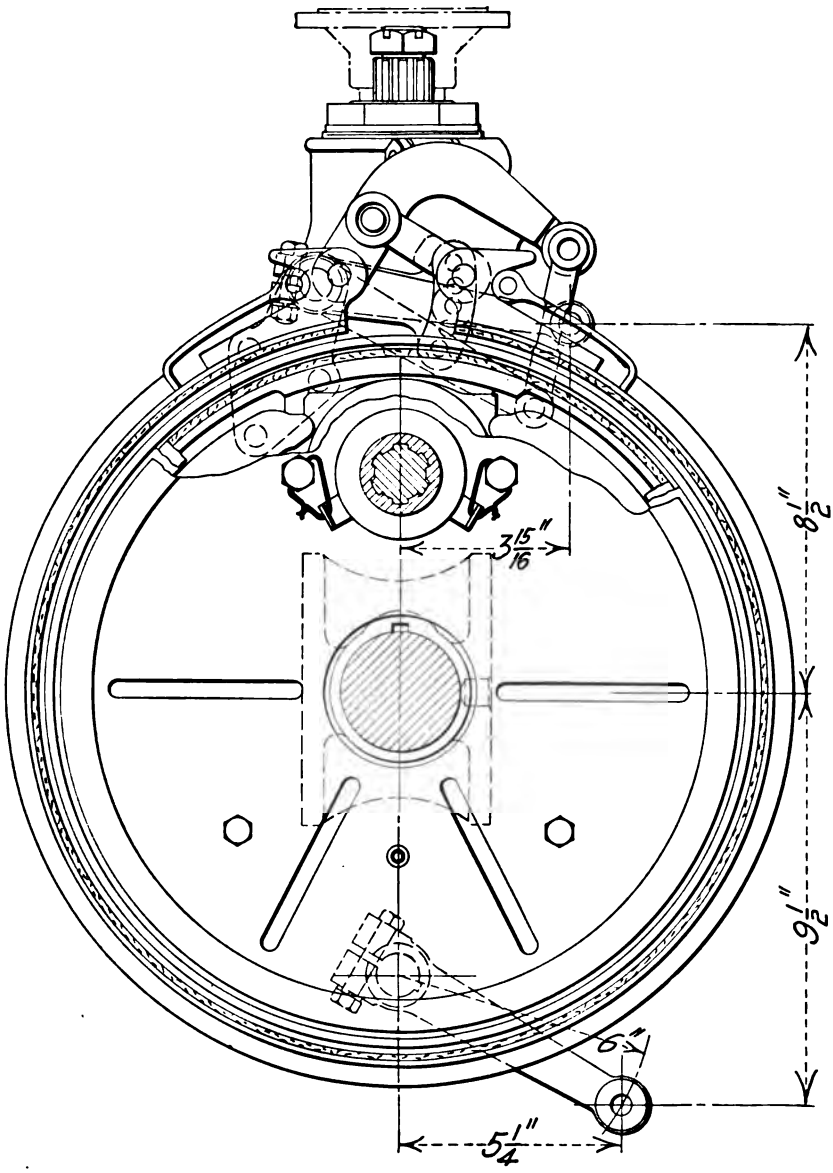


FIG. 359.—Russel internal gear drive axle.

The axle shown in (Figs. 358 and 359) has a normal capacity of about 2-tons; the maximum load on both spring-pads is 8,100 lb. which includes the portion of the weight of the chassis and body resting on these spring-pads. In this design there is no spacer between the inner races of the wheel bearings, but it embodies the construction adopted by the S.A.E. for front bearings, which eliminates this spacer.

A pressed steel cover *e* having a felt packing on its periphery is mounted within the brake drum to exclude dirt from the internal gear and to prevent the gear lubricant from reaching the brake band. It is attached to the brake support *g* by means of bolts, as shown, and is centered upon the wheel spacer *f*. A felt ring *h* is provided for the jackshaft outer bearing. The jackshaft pinion *i* may be removed without disturbing the jackshaft; formerly this pinion was riveted to the jackshaft end and this often necessitated a new jackshaft when a new pinion was required, because of the damage done when removing the pinion.

The pressed steel brake drum is made of ¼-in. stock 16½ in. outside diameter, and an overall width of 4⅜ in. To this brake drum is riveted the internal gear, (Fig. 360), made of 1035 S.A.E. steel, heat-treated, or 2320 carbonized. The details of the teeth are given in the drawing. Figure 361 is a detail of the jackshaft pinion, made of 2320 S.A.E. steel, carbonized. The pinion has 11 teeth and the internal gear 61, thus the ratio is $^{61}/_{11} = 5.54$ to 1.

Figure 362 is the jackshaft torque yoke shown at the right-hand side in the assembly; it is keyed to the jackshaft housing and attached to the brake support at *j*. This yoke carries the entire torque reaction, for at the left side the jackshaft housing is not rigidly attached. The spiral bevel gear drive pinion is made of nickel steel (S.A.E. 2320) carbonized. It has 10 splines where attached to the universal flange. This pinion has 15 teeth, and the drive gear, (Fig. 363), 24 teeth. The latter is riveted to the differential carrier as shown in the assembly by rivets *p*. Instead of fiber washers, as formerly used in this differential, the manufacturers now employ hardened and ground steel thrust washers, *g*. The differential side gears are carried in a heat-treated spider. It is claimed that in the event of any deflection of the dead axle which may be transmitted through the jackshaft, the stress produced will be taken through the

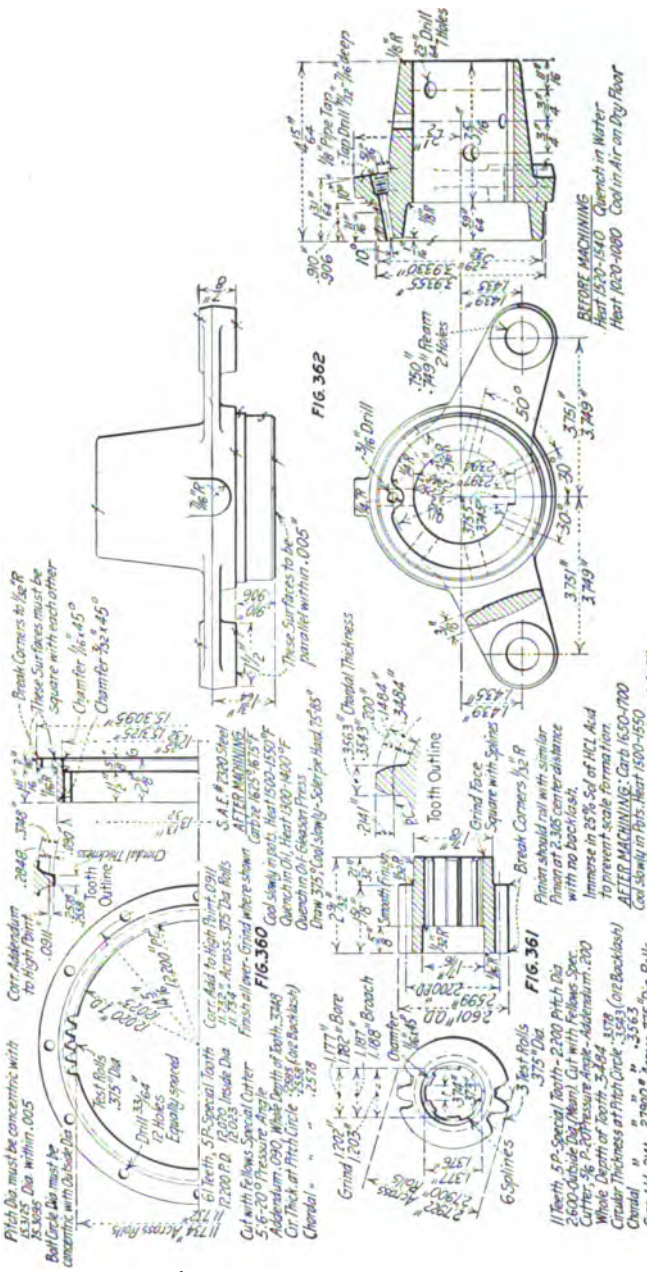


FIG. 362

FIG. 360

FIG. 361

FIG. 362.—Jackshaft torque yoke (Russel axle).

FIG. 360.—Internal gear. FIG. 361.—Jackshaft pinion.

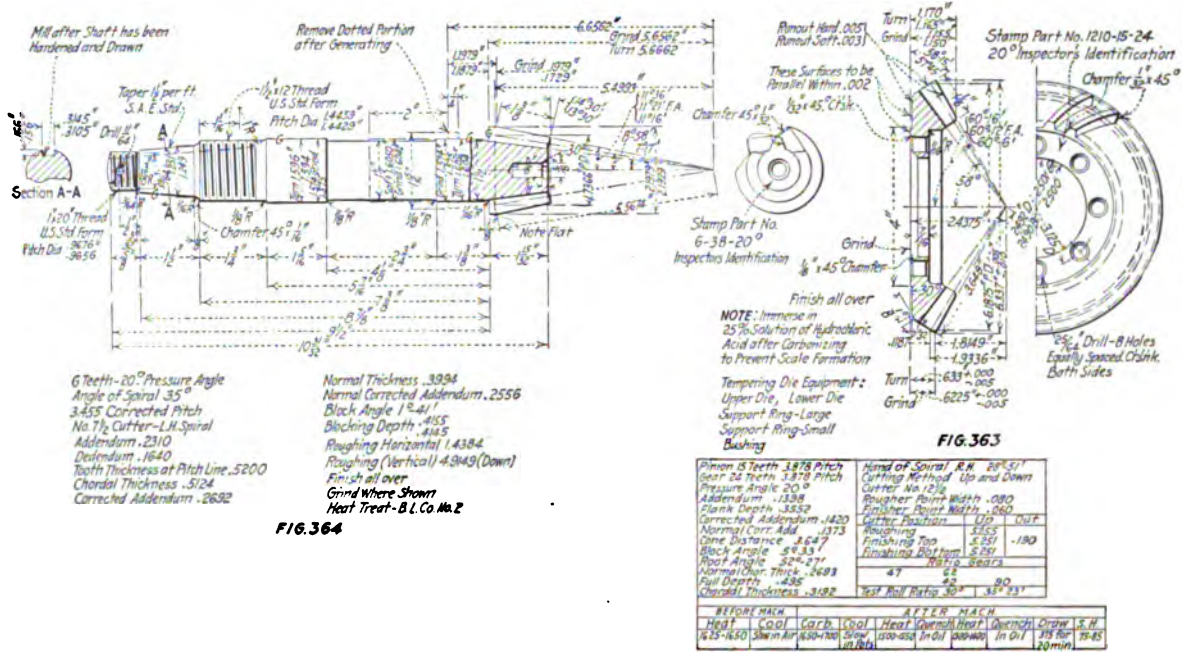

Mill after Shaft has been Hardened and Drawn
Remove Dotted Portion after Generating
Section A-A
Taper ½" per ft. S.A.E. Std.
Drill ¹¹/₆₄"
Chamfer 45° 1/16"
Note Flat
6 Teeth-20° Pressure Angle
Angle of Spiral 35°
3.455 Corrected Pitch
No. 7½ Cutter-L.H. Spiral
Addendum .2310
Dedendum .1640
Tooth Thickness at Pitch Line .5200
Chordal Thickness .5124
Corrected Addendum .2692
Normal Thickness .3994
Normal Corrected Addendum .2556
Blank Angle 1°41'
Blocking Depth .4155
Roughing Horizontal 1.4384
Roughing (Vertical) 4.9149 (Down)
Finish all over
Grind Where Shown
Heat Treat-B.L.Co. No. 2
FIG. 364
Pinout Hard .0051
Pinout Soft .0031
These Surfaces to be Parallel Within .002
Chamfer 45° 1/32" x 45° Chk.
Stamp Part No. 6-38-20° Inspectors Identification
Finish all over
NOTE: Immerse in 25% Solution of Hydrochloric Acid after Carbonizing to Prevent Scale Formation
Tempering Die Equipment: Upper Die, Lower Die
Support Ring-Large
Support Ring-Small
Bushing
Stamp Part No. 1210-15-24 20° Inspector's Identification
Chamfer 1/32" x 45°
Drill-8 Holes Equally Spaced Chkd. Both Sides
FIG. 363
Pinion 15 Teeth 3.978 Pitch
Gear 22 Teeth 3.978 Pitch
Pressure Angle 20°
Addendum .1398
Flank Depth .3552
Corrected Addendum .1420
Normal Corr. Add. .1373
Cone Distance 3.647
Blank Angle 5° 33'
Root Angle 32° 27'
Normal Chor. Thick. .2693
Full Depth .4395
Chordal Thickness .3192
Hand of Spiral R.H. 26° 51'
Cutting Method Up and Down
Cutter No. 12½
Roughing Point Width .080
Finisher Point Width .060
Cutter Position Up Out
Roughing .5565
Finishing Top .5251 .190
Finishing Bottom .5251
Ratio Gears
47 22
49 20 90
Test Roll Ratio 30° 33° 33'
BEFORE HACH
AFTER HACH
Heat Cool Carb. Cool Heat Quench Heat Quench Draw S.H.
1625-1650 Slow in Air 1650-1700 Slow in Bath 1500-1550 In Oil 1400-1450 In Oil 375 for 20 min. 75-85

FIG. 363.—Driving gear (Russel 2-ton axle). FIG. 364.—Spiral bevel pinion and shaft (Russel 1-ton pneumatic tired truck).

said spider and transferred to the differential case which has a larger diameter, and is therefore stronger than the hub of the side gear. The Russel Co. formerly used snap rings for holding the double row ball bearing, which takes the thrust load in addition to a portion to the radial load of the differential case, while at the present time nuts *n* are employed for clamping the ball races, with means for locking the nuts. While snap rings, no doubt, are attractive from the viewpoint of economy and simplicity, the manufacturers found that it required extreme accuracy in the width and location of the grooves, and the removal of the rings was very difficult. The double row bearing supporting the drive pinion was similarly mounted, but, for the same reasons, it is now clamped by nuts.

The differential is composed of two side gears *r*, and four pinions *s*. The former are made of 2320 S.A.E. nickel steel, and the latter of 1020 bar stock, both being properly carbonized and heat-treated. All the differential gears are carried on the spider *v* as a unit, and have a face width of $\frac{7}{8}$ in. The spider is a drop forging of 1025 stock, carbonized. The clearance for the running fit, of the pinion gears on the spider shanks when considering the tolerances, is from .002 to .006 in.

Figure 359 shows the means provided for adjusting the external brake. In all the important parts of the brake mechanism, castellated nuts are now used (together with pins or lock wire) instead of lockwashers as formerly. Dies are employed for handling the various gears in quenching during the heat-treatment, to prevent warping of the parts.

Figure 365 is an illustration of the 1-ton bevel drive axle for pneumatic tired trucks. Ball bearings are used throughout, new departure double row bearings being provided in the wheels and at the front end of the drive pinion shaft for taking the thrust load in both directions. This axle is built for a truck speed of from 25 to 35 miles per hour. The housing is made of pressed steel $\frac{3}{16}$ in. thick, with chrome-nickel steel reinforcing tubes, having a wall thickness of $\frac{5}{16}$ in. The brake drums are 18 in. in diameter. The internal brakes are of the full wrapping type, actuated by toggles. The drive shaft is made of chrome-nickel steel, having 10 splines at each end (instead of 6 as shown). Instead of retaining the wheel bearing by a threaded ring *a* clamped in place, as shown, this is now accomplished by a bearing retainer having a flange bolted to the brake support.

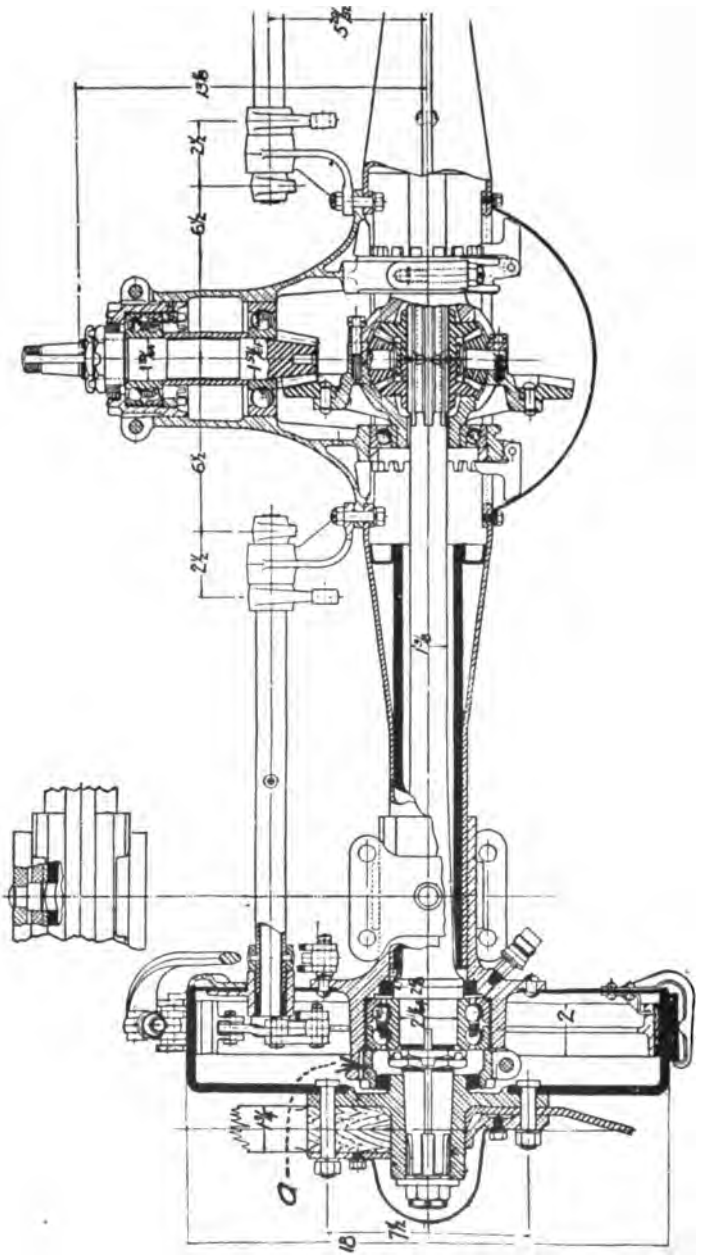


FIG. 365.—Russel 1-ton bevel drive rear axle for truck equipped with pneumatic tires.

Figure 364 is a detail drawing of the spiral bevel pinion and pinion shaft, giving full details of the teeth and the shaft.

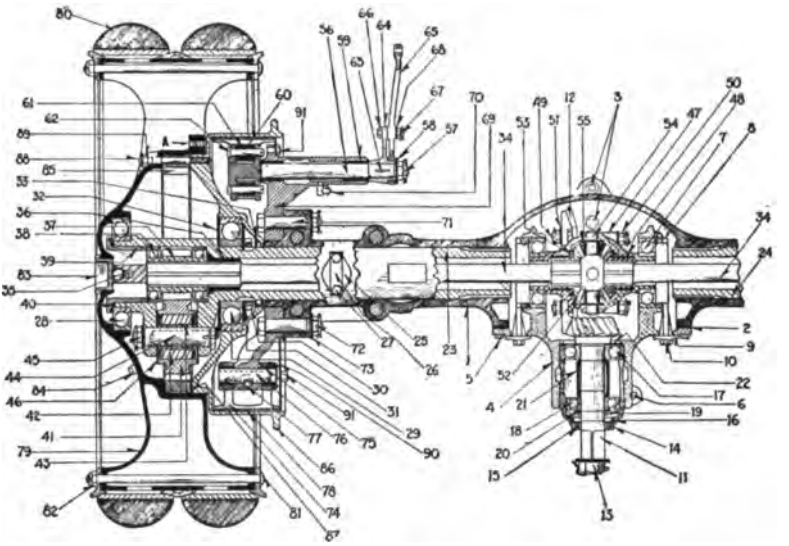


FIG. 366.—White 3½- and 5-ton internal gear drive axle (full-floating).

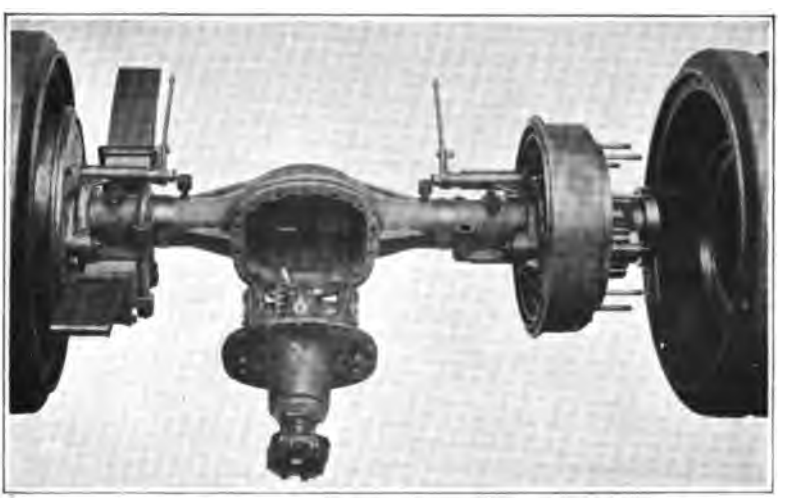


FIG. 367.—White 3½- and 5-ton truck rear axle.

Note that this pinion has only six teeth, but a spiral angle of 35°.

Figure 366 is the construction of the 3½- and 5-ton internal gear-driven axle of The White Co., (Fig. 367) being a photograph of this axle. In their 3½-ton and 5-ton axles, this firm uses one internal brake at the rear wheels, as the emergency brake, and a propeller shaft service brake foot-operated. The brake drum of the latter, being located about midway between the unit power plant and the rear axle, serves also for the purpose of supporting the weight of the propeller shaft at the center; a universal joint is located at the side of the brake drum. In this axle the first reduction is by means of spiral bevel gears, in the center of the axle, the second reduction by means of pinion 37 which is supported by two ball bearings 38 and is splined to the axle or drive shaft 34; this pinion meshes with an idler or intermediate gear 41 (running on Hyatt roller bearings), which in turn meshes with the wheel ring gear 85. This ring gear is plainly visible in the photograph. The joint between the wheel and axle housing is near the housing center at 33, hence the gears can run in a bath of oil with less danger of leakage than if there were a joint at, or near, the periphery of the drum.

REAR AXLE HOUSING AND BRAKE ASSEMBLY

Ref. No.	Number Per Car	Name
1	1	Rear axle housing.
	2	Rear axle housing grease oilers.
2	1	Rear axle housing gasket.
3	1	Rear axle housing pipe plugs (¾-in.).
4	1	Rear axle housing shaft case.
5	14	Rear axle housing shaft case bolts.
	14	Rear axle housing shaft case bolt lock washers.
6	1	Rear axle housing shaft case binder bolt (⅜ × 4 7/16-in.).
	1	Rear axle housing shaft case binder bolt nut (⅜-in.).
7	2	Rear axle housing shaft case bearings.
8	2	Rear axle housing shaft case bearing lock nuts.
	1	Rear axle thrust bearing nut.
9	2	Rear axle housing shaft case bearing lock nut locks.
10	4	Rear axle housing shaft case bearing lock nut screws.
11	1	Rear axle housing bevel gear pinion.
12	1	Rear axle housing bevel driving gear.
13	1	Rear axle housing bevel driving gear nut.
14	1	Rear axle housing bevel gear cage.
15	1	Rear axle housing bevel gear cage felt.
16	1	Rear axle housing bevel gear cage washer.
17	1	Rear axle housing bevel gear cage bearing.
18	1	Rear axle housing bevel gear cage bearing.
19	1	Rear axle housing bevel gear cage bearing lock nut.
20	1	Rear axle housing bevel gear cage bearing lock nut lock washer.
21	1	Rear axle housing bevel gear cage bearing spacer.
22	1	Rear axle housing bevel gear cage bearing shim (1/64-in.–1/32-in.).
23	1	Rear axle housing tube, R.H.
24	1	Rear axle housing tube, L.H.

Ref. No.	Number Per Car	Name
25	2	Rear axle housing tube locks.
26	2	Rear axle housing tube lock plates.
27	4	Rear axle housing tube lock plate screws (½-in.–18).
28	2	Rear axle housing tube bearings.
29	2	Rear axle housing tube bearings.
30	2	Rear axle housing tube bearing lock nuts.
31	2	Rear axle bearing washer.
32	1	Rear wheel inner bearing washer, R.H.
	1	Rear wheel inner bearing washer, L.H.
33	2	Rear axle housing tube bearing lock nut retainers.
	2	Rear axle housing tube bearing lock nut retainer felts.
34	2	Rear axle housing tube spindles.
35	2	Rear axle housing tube spindle steel balls.
36	2	Rear axle housing tube spindle bushings.
37	2	Rear axle housing tube spindle driving pinions.
38	4	Rear axle housing tube spindle driving pinion bearings.
39	2	Rear axle housing tube spindle driving pinion bearing nuts.
40	2	Rear axle housing tube spindle driving pinion bearing lock washers.
41	2	Rear axle housing tube intermediate gears.
42	2	Rear axle housing tube intermediate gear roller bearings (Hyatt high duty).
43	2	Rear axle housing tube intermediate gear spacers.
44	2	Rear axle housing tube intermediate gear pins (1⅛ in.–20).
45	2	Rear axle housing tube intermediate gear pin nuts.
46	4	Rear axle housing tube intermediate gear washers.
47	1	Rear axle differential case, R.H.
	1	Rear axle differential case, L.H.
48	2	Rear axle differential case bushings.
49	4	Rear axle differential case bolts.
50	4	Rear axle differential case bolt nuts.
51	8	Rear axle differential case bolt lock washers.
52	2	Rear axle differential case gears.
53	2	Rear axle differential case gear washers.
	2	Rear axle differential case gear washers.
54	1	Rear axle differential case spider.
55	4	Rear axle differential case spider pinion gears.
56	2	Rear axle brake operating shafts.
57	2	Rear axle brake operating shaft nuts.
58	2	Rear axle brake operating shaft nut washers.
59	4	Rear axle brake operating shaft bushings.
60	4	Rear axle brake operating shaft links.
61	8	Rear axle brake operating shaft link bushings.
62	4	Rear axle brake operating shaft link pins (short).
	4	Rear axle brake operating shaft link pins (long).
63	2	Rear axle brake operating shaft keys (No. 15 Woodruff).
64	2	Rear axle brake operating shaft levers.
65	2	Rear axle brake operating shaft adjusting levers.
66	2	Rear axle brake operating shaft adjusting lever bolts.
67	2	Rear axle brake operating shaft adjusting lever bolt nuts.
68	2	Rear axle brake operating shaft adjusting lever bolt lock washers.
69	2	Rear axle brake support brackets.
70	2	Rear axle brake support bracket grease oilers.
71	4	Rear axle brake support bracket bolts.
72	4	Rear axle brake support bracket bolt nuts
73	4	Rear axle brake support bracket bolt nut washers.
74	4	Rear axle brake shoes.
	4	Rear axle brake shoe springs.
	4	Rear axle brake shoe spring pins.
75	8	Rear axle brake shoe bushings.

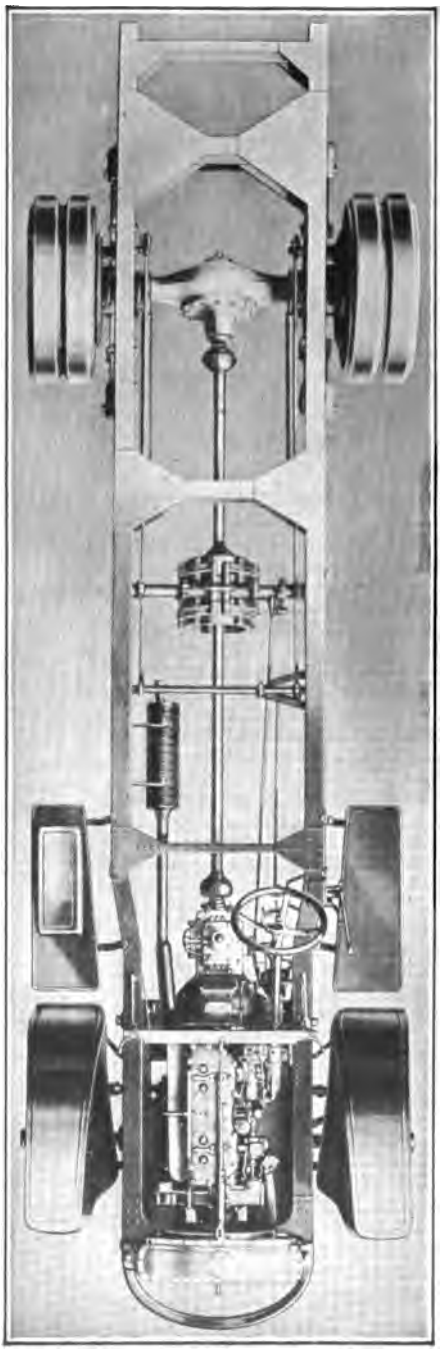


FIG. 368.—White 3½- and 5-ton truck chassis.

Ref. No.	Number Per Car	Name
	8	Rear axle brake shoe bushing dowels.
76	4	Rear axle brake shoe pins (½ in.–18).
77	2	Rear axle brake shoe wedge pins.
	2	Rear axle brake shoe wedge pin nuts.
78	4	Rear axle brake shoe linings.
	96	Rear axle brake shoe lining rivets (brass 5/16- × 1¾-in.).
79	2	Rear wheels.
80	2	Rear wheel tires (40 × 5 in. dual).
81		This part not used.
82		This part not used.
83	2	Rear wheel hub caps.
	2	Rear wheel hub cap gaskets.
84	2	Rear wheel oil drain plugs (¾-in. pipe).
85	2	Rear wheel ring gears.
86	2	Rear wheel drums and covers.
87	2	Rear wheel drum and cover gaskets.
88	24	Rear wheel drum and cover studs.
89	24	Rear wheel drum and cover stud nuts.
90	4	Rear wheel drum and cover plates.
91	6	Rear wheel drum and cover plate screws.
	4	Rear axle brake shoe spring pins.
	2	Rear wheel stud washers.
	2	Rear wheel drum and cover plate bolts.
	4	Rear wheel drum and cover plate bolt nuts.
	8	Rear wheel drum and cover plate bolt lock washers.

The wheels are full-floating, each wheel being supported by two ball bearings 28 and 29, placed a great distance apart, for the

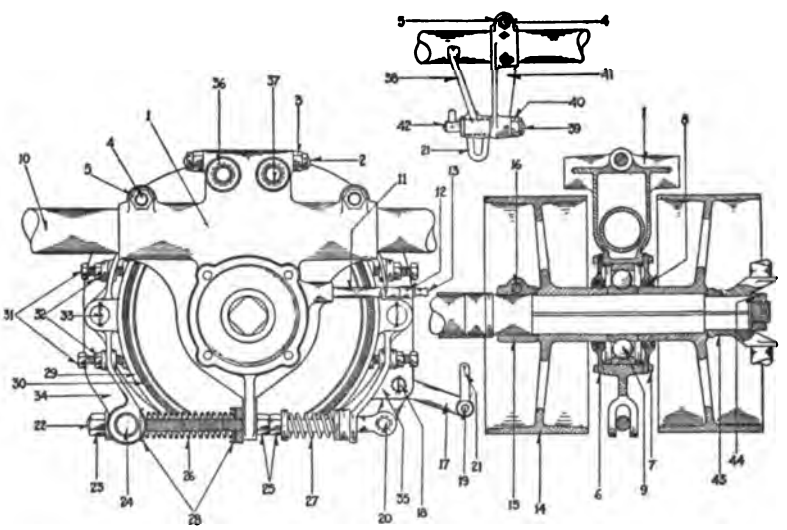


FIG. 369.—White truck propeller shaft brake.

purpose of reducing the bearing pressure (see page 334). Figure 368 is a plan view of the White 3½-ton and 5-ton truck. The foot

service brake, the propeller shafts, and the radius and thrust tubes can be seen.

The service brake consists of two propeller shaft brake drums 14 (see Fig. 369), which are held on the shaft by means of a square fitting, which are held in place by nut 44, which is immediately behind the brake drums, and within the universal joint, between the two halves of the propeller shaft. The bracket 1, which is a part of the frame cross-tube 10, supports the center of the propeller shaft and forms the brake anchorage. The ball bearing 9, between the two drums, supports the shaft. When applying the foot pedal, an upward pull is exerted on link 21, and this causes the brake shoes to contract upon the two drums. By referring to this figure as well as to the chassis plan view, the action of the brake will be apparent.

FOOT SERVICE BRAKE ASSEMBLY

Ref. No.	Number Per Car	Name
1	1	Drive shaft bracket.
2	2	Drive shaft bracket wedge pins.
3	2	Drive shaft bracket wedge pin nuts.
4	2	Drive shaft bracket binder bolts.
5	2	Drive shaft bracket binder bolt nuts.
6	2	Drive shaft bracket caps.
7	8	Drive shaft bracket cap screws.
	8	Drive shaft bracket cap screw lock washers.
8	2	Drive shaft bracket cap felts.
9	1	Drive shaft bracket bearing.
10	2	Drive shaft bracket support tube.
11	1	Drive shaft bracket grease tube (¼- × 6-in.).
12	1	Drive shaft bracket grease tube reducer.
13	1	Drive shaft bracket grease oiler.
14	2	Drive shaft brake drums.
15	1	Drive shaft brake drum lock nut.
16	1	Drive shaft brake drum binder bolt.
	1	Drive shaft brake drum binder bolt lock washer.
17	1	Drive shaft brake operating lever.
18	2	Drive shaft brake operating lever pin.
19	1	Drive shaft brake operating lever pin.
	1	Drive shaft brake operating lever bushing (long).
	1	Drive shaft brake operating lever bushing (short).
20	1	Drive shaft brake operating lever pin.
21	2	Drive shaft brake operating lever clevises.
22	1	Drive shaft brake adjusting cross rod.
23	1	Drive shaft brake adjusting cross rod nut.
24	1	Drive shaft brake adjusting cross rod swivel pin.
25	2	Drive shaft brake adjusting cross rod stop and check nuts.
26	1	Drive shaft brake release spring (long).
27	1	Drive shaft brake release spring (short).
28	4	Drive shaft brake release spring cups.
	2	Drive shaft brake release spring cup washers.
29	4	Brake shoes.
30	4	Brake shoe linings.
	40	Brake shoe lining rivets.

Ref. No.	Number Per Car	Names
31	8	Brake shoe adjusting screws.
32	8	Brake shoe adjusting screw nuts.
33	2	Brake shoe pins.
34	1	Brake shoe supporting arm, R.H.
35	1	Brake shoe supporting arm, L.H.
36	4	Brake shoe supporting arm bushings.
37	2	Brake shoe supporting arm pins.
38	1	Bell crank lever.
39	1	Bell crank lever pin.
40	1	Bell crank lever pin nut.
41	1	Bell crank bracket.
	1	Bell crank bracket binder bolt.
	1	Bell crank bracket binder bolt nut.
	1	Bell crank bracket bushing.
42	1	Bell crank bracket grease oiler.
43	1	Drive shaft trunnion.
44	1	Drive shaft trunnion nut.
	1	Drive shaft tube support spacer.
	1	Drive shaft tube support spacer bolt.
	1	Drive shaft tube support spacer bolt nut.

The emergency brake, within the rear wheel brake drums, is operated by the brake shaft 1, (Fig. 370), located in the front of

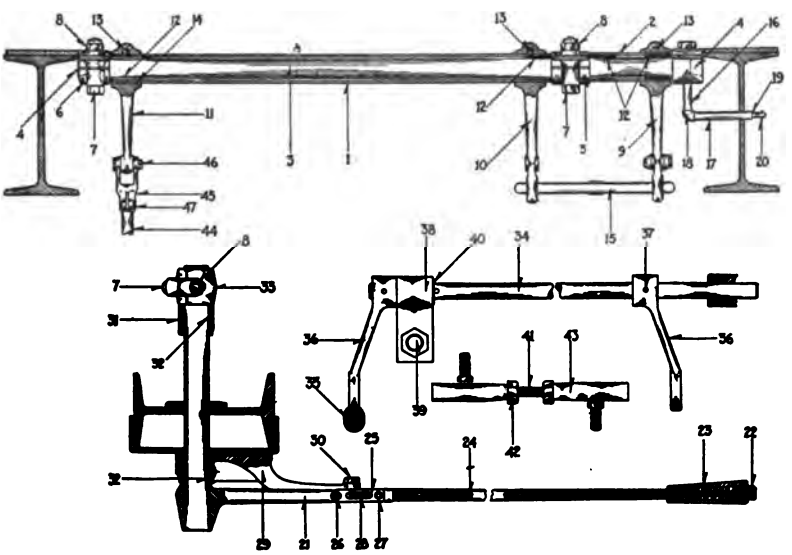


FIG. 370.—White truck brake shaft assembly.

the propeller shaft brake. Details of this brake shaft assembly can be seen from the drawing. By exerting a pull in the center of the brake shaft lever link 15, an equal force is exerted on both the levers 9 and 10, as the link 15 is not rigidly attached to these levers.

The brakes on both rear wheels are thus equalized, for should the brake on one side have more play than the other, the link 15, having levers 9 and 10 the same distance from its center, will exert an equal pull on them, and should one brake happen to have more play this play will be taken up (the lever which has more play will move farther than the other with less play), until the pressure on both is equalized.

Figure 370 also gives the emergency or hand brake lever assembly and the center view shows the carburetor control shaft, located in front of the foot pedals. The emergency brake lever is located on the outside; the left frame side members can be seen from the chassis plan view.

BRAKE SHAFT ASSEMBLY

Ref. No.	Number Per Car	Names
1	1	Brake shaft (outside, long)
2	1	Brake shaft (outside, short)
3	1	Brake shaft (inside).
4	2	Brake shaft plugs.
5	1	Brake shaft bracket (right hand).
6	1	Brake shaft bracket (left hand).
7	2	Brake shaft bracket bolts.
8	2	Brake shaft bracket bolt nuts.
	2	Brake shaft bracket bolt lock washers.
9	1	Brake shaft lever.
10	1	Brake shaft lever.
11	1	Brake shaft lever.
12	4	Brake shaft lever bushings.
13	3	Brake shaft lever binder bolts.
	3	Brake shaft lever binder bolt nuts.
14	1	Brake shaft lever key.
15	1	Brake shaft lever link.
16	1	Brake shaft grease nipple (short).
17	1	Brake shaft grease nipple (long).
18	1	Brake shaft grease nipple elbow.
19	1	Brake shaft grease nipple coupling
20	1	Brake shaft grease oiler.
21	1	Hand brake lever.
22	1	Hand brake lever release plunger.
23	1	Hand brake lever release plunger spring.
24	1	Hand brake lever latch rod.
25	1	Hand brake lever dog.
26	2	Hand brake lever dog bolts.
27	2	Hand brake lever dog bolt nuts.
28	1	Hand brake lever quadrant.
	2	Hand brake lever quadrant bolts.
29	1	Hand brake lever quadrant bracket.
30	2	Hand brake lever quadrant bracket bolts.
31	1	Hand brake lever bracket.
32	2	Hand brake lever bracket bushings.
33	1	Hand brake lever shaft lever.
	1	Hand brake lever shaft lever rod (front, $^{17}/_{32}$- × 69¼-in.).
	2	Hand brake lever shaft lever rod ends.
	2	Hand brake lever shaft lever rod end pins.

Ref. No.	Number Per Car	Names
34	1	Carburetor control shaft.
35	1	Carburetor control shaft spring.
36	2	Carburetor control shaft levers.
37	2	Carburetor control shaft lever taper pins.
38	1	Carburetor control shaft lever bracket.
39	1	Carburetor control shaft bracket screw.
40	1	Carburetor control shaft bracket screw washer.
41	1	Carburetor control rod.
42	2	Carburetor control rod nuts.
43	2	Carburetor control rod ball joints.
44	2	Hand brake rods ($1\frac{7}{32}$- × $61\frac{1}{4}$ in., rear).
45	4	Hand brake rod ends.
46	4	Hand brake rod end pins.
47	4	Hand brake rod end binder screws.
	2	Hand brake lever shaft lever rod end binder screws.

Figure 371 illustrates the two halves of the propeller shaft or drive shaft (the upper view the rear half, the lower, the front half). In the standard 3½-ton truck, the length of the rear half

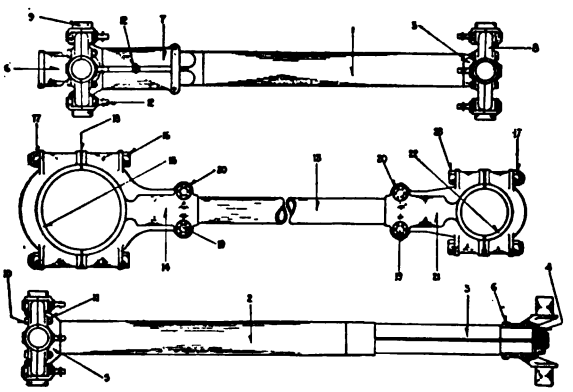


FIG. 371.—White truck propeller shaft and radius tube.

is 34½ in. and the front half almost 50 in. The center view illustrates one of the radius tubes, their location being visible from the chassis view.

DRIVE SHAFT AND RADIUS TUBE ASSEMBLY

Ref. No.	Number Per Car	Name
1	1	Drive shaft (brake to rear axle).
2	1	Drive shaft (transmission to brake).
3	1	Drive shaft extension ($16\frac{13}{16}$ in. long, through service brake).
4	1	Drive shaft nut.
5	2	Drive shaft trunnions.
6	2	Drive shaft trunnions.
7	1	Drive shaft sliding box.

Ref. No.	Number Per Car	Name
8	3	Drive shaft universal joint rings (6 halves).
9	12	Drive shaft universal joint ring bushings.
	12	Drive shaft universal joint ring dowels.
10	24	Drive shaft universal joint ring bolts.
11	24	Drive shaft universal joint ring bolt nuts.
12	2	Drive shaft universal joint box grease oilers.
12	8	Drive shaft universal joint ring grease oilers.
	3	Drive shaft universal joint ring boots.
13	2	Radius tubes.
14	2	Radius tube ends (large).
15	2	Radius tube end bushings (large).
16	4	Radius tube end bolts.
17	4	Radius tube end bolt nuts.
18	8	Radius tube end shims ($\frac{1}{64}$-in., $\frac{1}{32}$-in., $\frac{1}{16}$-in., $\frac{1}{8}$-in. thicknesses).
19	8	Radius tube end binder bolts.
20	8	Radius tube end binder bolt nuts.
21	2	Radius tube ends (small).
22	2	Radius tube end bushings.
23	4	Radius tube end bolts.
	4	Radius tube end bolt nuts.
	2	Radius tube balls.

The internal gear drive axle used on the 1-ton truck manufactured by the International Harvester Corp. (see Figs. 372 and 372*a*) has a forged chrome-nickel steel load carrying member, *L* of round section, like the Russell, except that it is flattened in the center and curved to make room for the differential housing. The latter is bolted to the differential carrier as shown in the figure. The driving shafts *D* are exposed between the differential housing and the brake support castings which contain the spur gear pinion. The internal gear is enclosed by this brake support casting to which are fitted the two internal expanding brakes, the latter being cam operated. All the bearings are of the Hyatt roller type, with the exception of a double row ball bearing for the internal spur gear pinion and thrust ball bearings to take the thrust, in both directions, of the bevel pinion shaft, and a thrust bearing behind the bevel drive gear to take the thrust arising from the bevel ring gear. The lateral thrust against the wheels in either direction is taken by the two sets of thrust washers in each hub, composed of one bronze washer between two steel washers. By having the bronze washers bear against steel, a good wearing surface is obtained. The outer set of washers *B* comes into action when the thrust is against the

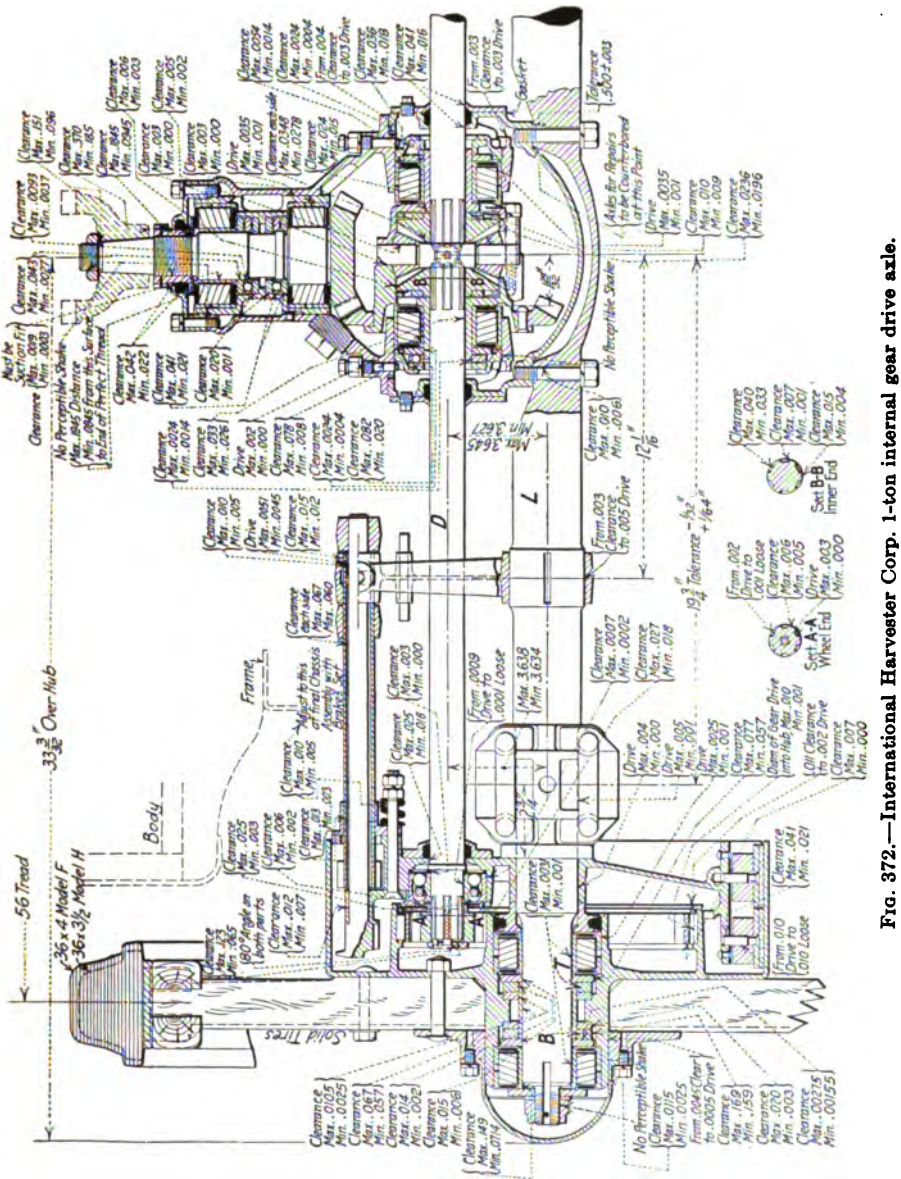


FIG. 372.—International Harvester Corp. 1-ton internal gear drive axle.

inner side of the wheel, and when the thrust is from the outside (when hitting the curb, for instance) the inner washers *T* will resist the thrust load.

All the Hyatt bearings in these axles are of the same size, which

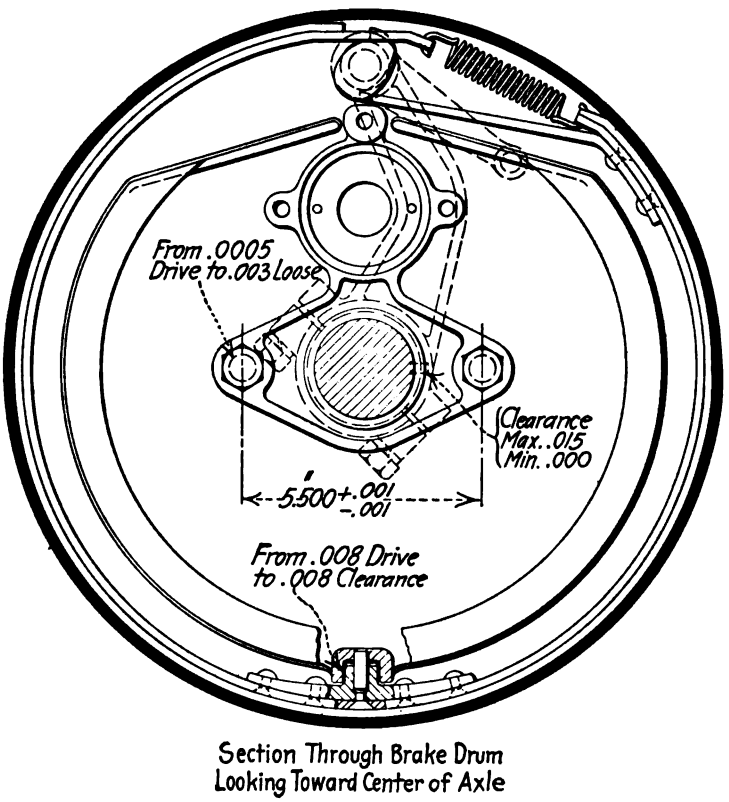


FIG. 372*a*.—International Harvester Corp. truck brake.

is an advantage when repairs are necessary. The clearances and the drive fits of the various parts are noted on the drawing and should prove useful to students. Where the driving fit is given, it means that the bore is made smaller by an amount equal to the dimensions given, for the purpose of a drive fit. Sometimes where great accuracy is not of vital importance the two parts may have a tolerance of several thousandths of an inch clearance to a driving fit of .005 or more.

Figure 373, gives a general view of the 1- to 1½-ton truck, Walker-Weiss axle, built by the Flint Motor Axle Co. The first reduction is obtained in the center of the axle by means of a pair of bevel gears; the other reduction takes place at the outside of each wheel by means of a stationary ring gear, and two or three pinions attached to the wheel driving cap. This is said to balance the drive and reduce the size of the gears required. By having a plurality of pinions the tooth pressure is small compared with the single pinion drive, hence smaller gears may be employed. The main shaft made of chrome-nickel steel has at its outer end a gear pinion which is in mesh with two or three planetary or idler gears and these are in mesh with the stationary ring gear. When the shaft rotates the pinion *d* at its outer end drives the idler gears *e*. The ring gear *r* being stationary, takes the reaction from the idler gears. In this manner, these idler gears, which are studded to the wheel driving flange *f*, rotate about the studs, and since the ring gear is stationary, the driving flange will be rotated in the same direction as the drive shaft. The axle is of the full-floating type. The housing which covers the driving shaft serves as the load carrying member. The wheel bearings, the driving pinion and idler pinions, run in a bath of oil. As in the full-floating axle in general, the gears may be replaced without jacking up the truck or removing the wheels. In this construction it is claimed that the internal gears are protected absolutely from dust and grit and the oil flows from one wheel to the other through the axle housing and the differential.

The Clark axle, (Fig. 374), employs a round load carrying member, bent in the center to clear the differential housing. In this axle the drive shaft is in front of the load carrying member. The brake drum contains a partition to separate it from the internal gear but there is only a small amount of space between them. Hyatt roller bearings are employed to carry most of the radial load while the double row ball bearings at the outer ends of the axle and at the pinion shaft are used to carry a portion of the radial load and to take the thrust in both directions; thrust ball bearings are installed behind the differential housing. The internal gear is attached to the hub by bolts passing through the gear, the brake drum, and the wheel spokes.

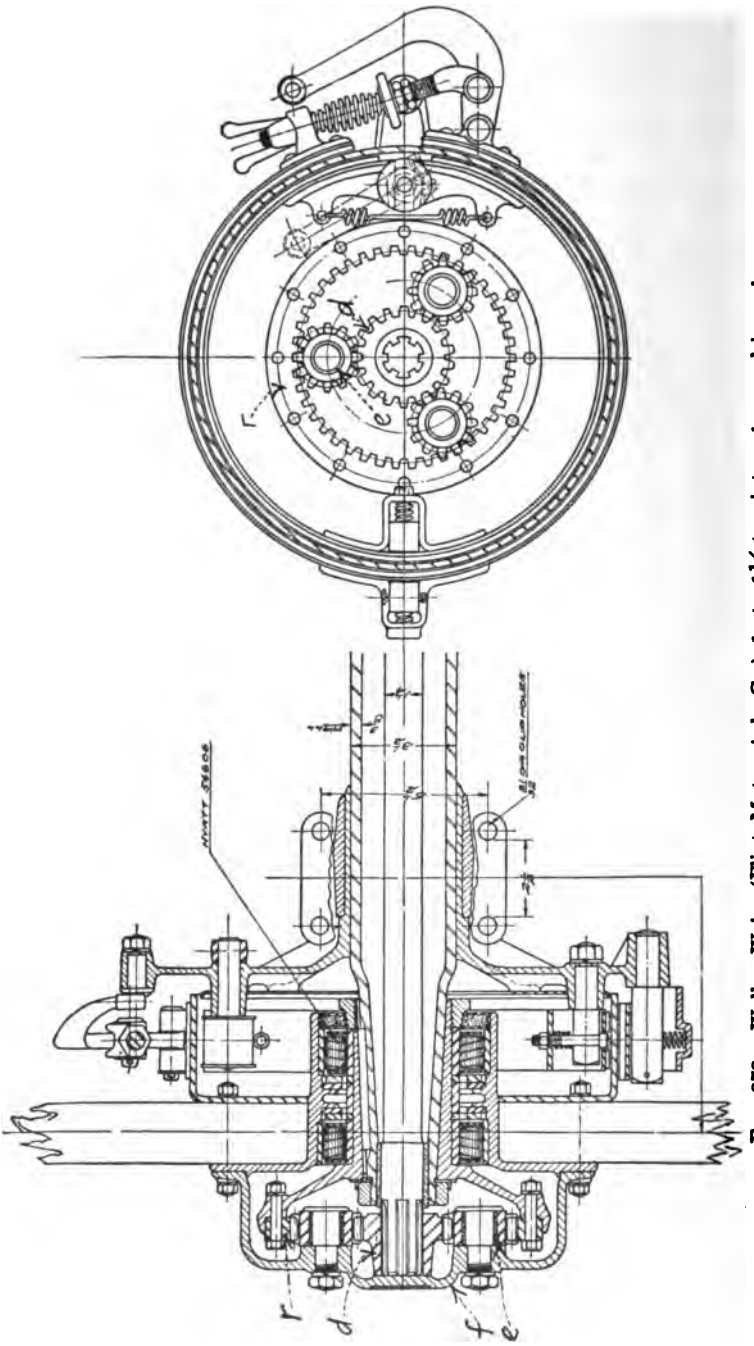


FIG. 373.—Walker-Weiss (Flint Motor Axle Co.) 1- to 1½-ton internal gear drive axle.

FIG. 374.—Clark internal gear drive axle.

The Double Reduction Axle.—The term double reduction axle as employed to-day denotes a two-fold reduction in the center of the rear axle housing. (In reality, the chain drive as well as the internal gear drive, are double reduction axles.) In some of the double reduction axles, the first reduction is accomplished by spur gears or spiral spur gears, as in the White 1½- to 2-ton axle, and the second reduction by means of bevel gears; while in others, like the Autocar, and the small Mack truck, the first reduction is by bevel gears and the second by spur

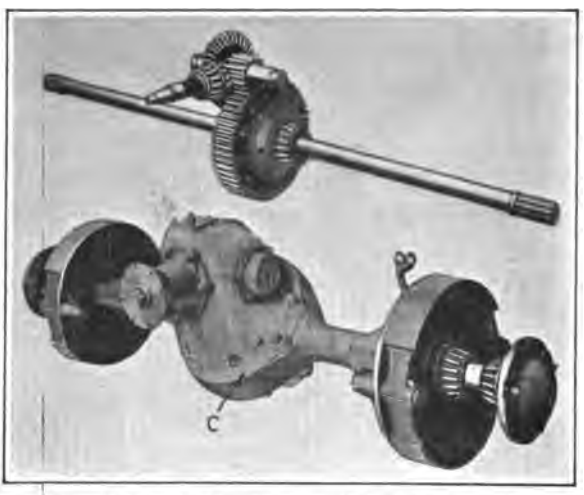


FIG. 375.—Mack double-reduction 1½- to 2-ton truck rear axle.

gears. The last named truck, manufactured by The International Motor Co. (manufacturers of the Mack truck) which used a worm drive, as well as a chain drive, on its 1½- to 2-ton model, since 1914, has recently abandoned the worm drive altogether and substituted therefore a double reduction axle, for which several important advantages are claimed. It provides 10¼-in. road clearance as against 9½ in. with the worm drive. Figure 375 shows two views; at the top, the double reduction gears and the axle shafts, and at the bottom, the complete rear axle. The main axle member (Fig. 376), is a drop-forged banjo or yoke with hollow tubular ends carrying the axle shafts, as in the conventional live axle, with the driving members inclosed within the banjo yoke. The drive from the propeller shaft is in a straight line, passing first through a pair of bevels and then through a pair of spur gears to the differential.

The banjo is inclined at an angle of 45°, to be in a better position to resist road shocks which are normally oblique in direction. By removal of the aluminum cover at the back, the differential and bull gear *G* (see also Fig. 379) may be taken out bodily while the aluminum gear carrier *C* at the front, may be removed entire which permits the pinion shaft or jackshaft to be removed individually without disturbing other portions of the assembly.

Four different gear ratios are provided on this axle, 5⅞, 7$^{6}/_{10}$, 9¼, and 10¾, the design of the carrier making it possible to effect a change of ratio by drawing out the jackshaft and pinion shaft and substituting different sized bevel gears.

Figure 377 shows a cross-section through the rear wheel while Fig. 379 is a section through the double reduction, including the

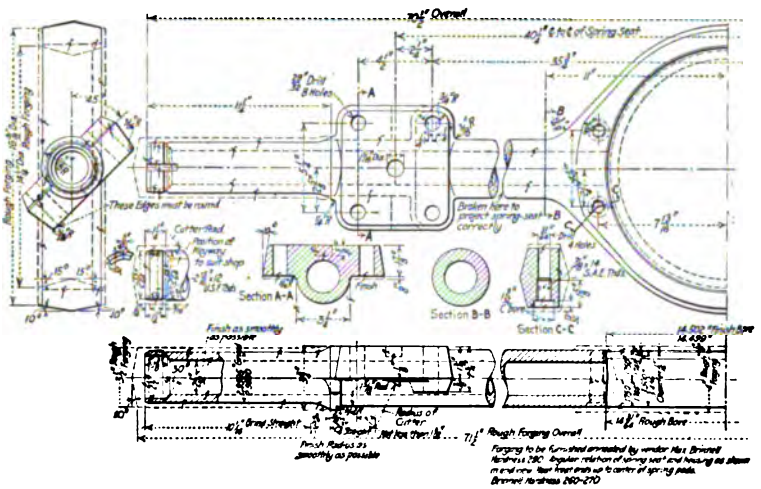
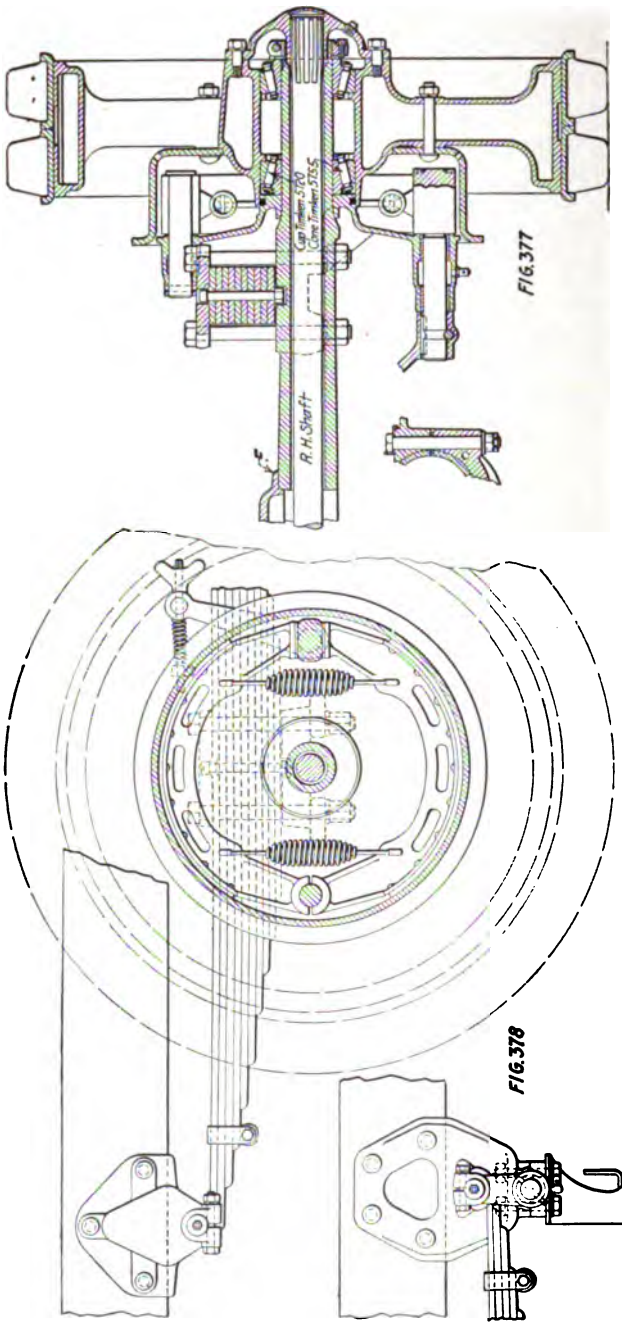


FIG. 376.—Mack double-reduction rear axle housing (drop forging).

differential. The flange *F* being part of the housing containing the bevel and spur gears. The pinion is made integral with the rear axle pinion shaft *B*; the latter drives the bevel gear *C* keyed to the jackshaft which is made integral with the spur gear before mentioned. This spur gear meshes with the big spur or bull gear *G*. All bearings in this axle are of the Timken taper roller type. The brake shafts have self-lubricating bushings. The axle shafts are splined into the differential hubs and the drop-forged hub caps; the construction is of the full-floating type. The plate at the top of the carrier has a breather to relieve pressure within.



FIGS. 377 and 378.—Mack double-reduction rear axle (cross-section through wheel and brake mechanism).

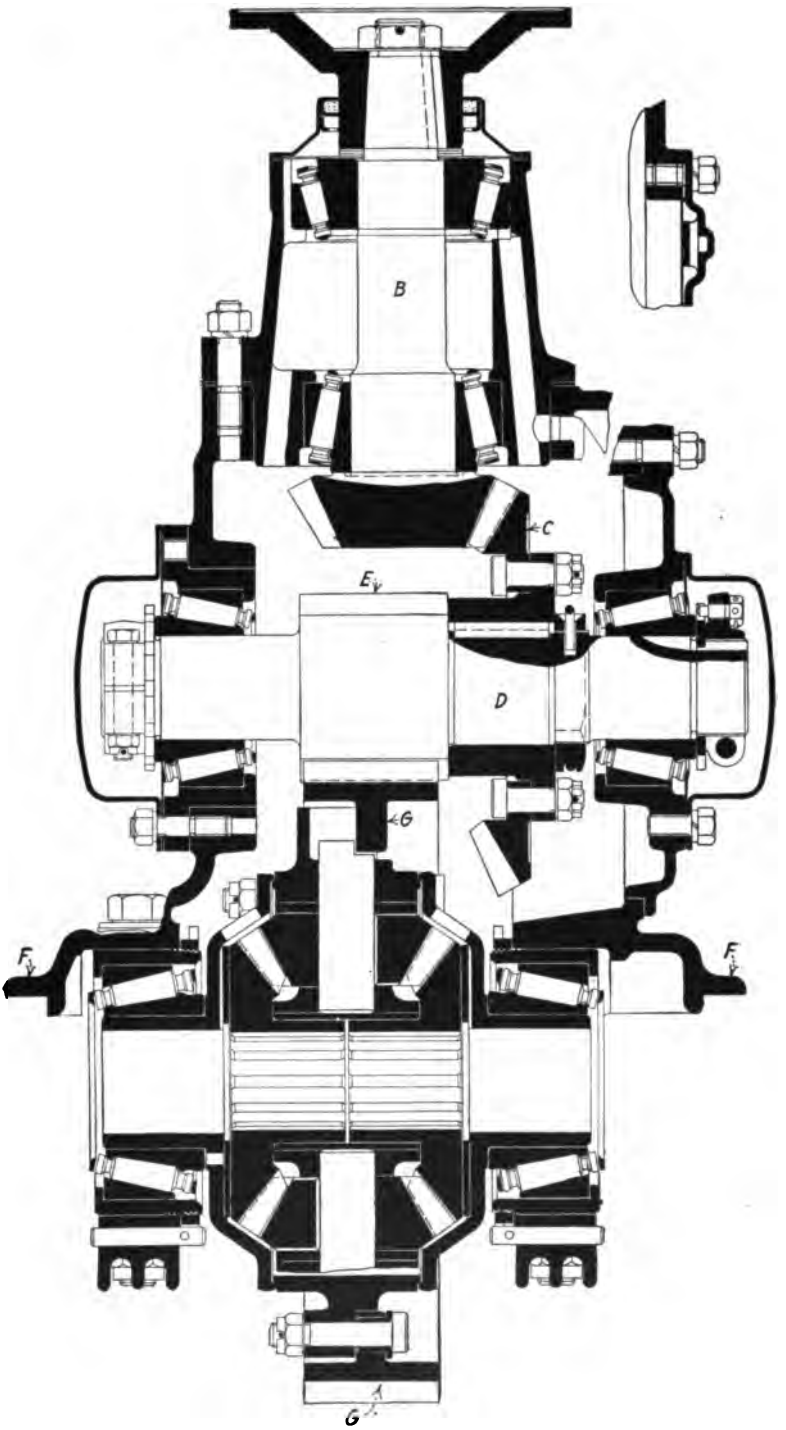


FIG. 379.—Cross-section through double-reduction Mack rear axle.

Brakes.—The hand brake, mounted mid-length on the drive shaft is 11 in. in diameter and 6 in. wide. It is of the contracting type, the shoes being deeply ribbed drop-forgings, and the entire brake assembly mounted on a rigid three-point-supported drop-forged frame between two closely spaced cross-members as seen from Fig. 4. Each lining is 6 by 12 in. in size.

Figure 378 shows the brake mechanism and the front and rear

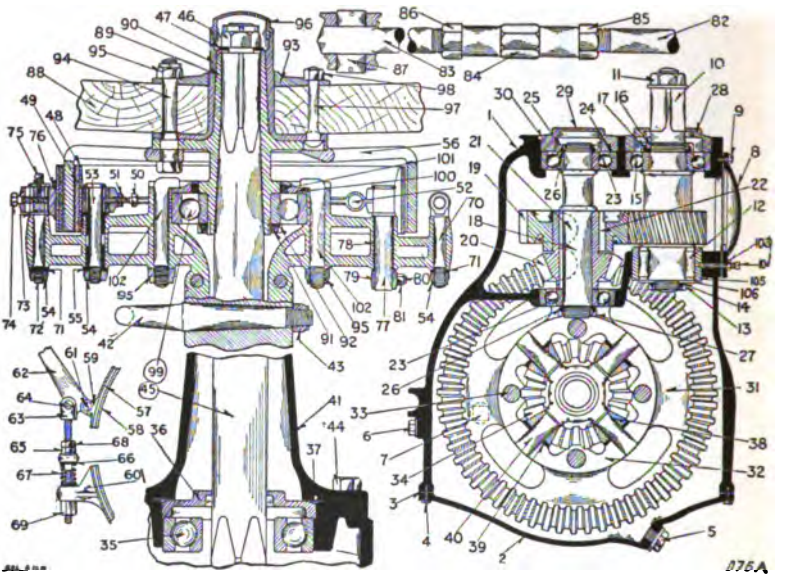


FIG. 380.—White 1½- to 2-ton double-reduction truck axle. (See page 312 for names of parts.)

spring brackets. The front end being anchored while the rear is shackled.

The advantages claimed for the double reduction are that its efficiency is constant throughout speed and load ranges and wear, that it is more durable than the worm and the internal gear drive, and, like the worm drive, it is perfectly enclosed.

Another example of a double reduction model is the White 1½- to 2-ton truck axle, an assembly of which is shown in Fig. 380. The first reduction is obtained by spiral spur gears, the drive coming through the pinion countershaft 10, with which the spur gear is made integral, the latter meshing with spiral spur gear 19.

This gear is keyed to the hub of the bevel pinion 20, which meshes with the bevel driving gear 31, which forms the second reduction. The axle or drive shaft 45 is semi-floating, the wheel being supported by a ball bearing 99. The names of the different parts as used by the White Co., are given in the table for reference.

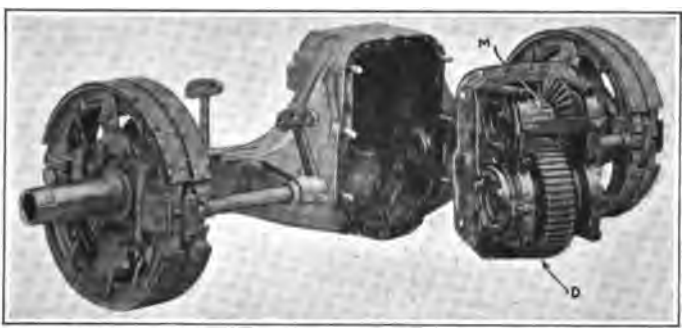


FIG. 381.—Autocar double-reduction rear axle.

Figures 381 and 382 show the double reduction rear axle used on the Autocar truck. In this axle the first reduction is by bevel

FIG. 382.—Autocar double-reduction rear axle.

gears, the second by spur gears. All the gears are assembled on the front cover, shown in Fig. 382, while, for inspection and adjustment, a cover plate can be removed from the back of the axle housing.

Ref. No.	Number Per Car	Name
1	1	Rear axle gear case.
2	1	Rear axle gear case cover.
3	1	Rear axle gear case gasket.
4	11	Rear axle gear case screws.
	11	Rear axle gear case screw lock washers.
	3	Rear axle gear case screw dowels.
5	1	Rear axle gear case filler hole plug.
	1	Rear axle gear case ratio plate.
	2	Rear axle gear case ratio plate screws.
6	1	Rear axle gear case drain plug.
7	1	Rear axle gear case overflow plug.
	1	Rear axle gear case overflow plug washer.
8	1	Rear axle gear case hand hole cover.
9	5	Rear axle gear case hand hole cover screws.
	5	Rear axle gear case hand hole cover lock washers.
10	1	Rear axle gear case countershaft and gear.
11	1	Rear axle gear case countershaft nut.
12	1	Rear axle gear case countershaft bearing (rear).
13	1	Rear axle gear case countershaft bearing lock nut.
14	1	Rear axle gear case countershaft bearing lock nut washer.
15	1	Rear axle gear case countershaft bearing (front).
16	1	Rear axle gear case countershaft lock nut, L.H.
17	1	Rear axle gear case countershaft lock nut lock washer.
18	1	Rear axle gear case pinion shaft.
19	1	Rear axle gear case spiral gear.
20	1	Rear axle gear case pinion shaft gear.
21	2	Rear axle gear case pinion gear keys.
	1	Rear axle gear case pinion shaft gear lock nut.
	1	Rear axle gear case pinion shaft gear lock washer.
22	1	Rear axle gear case pinion shaft spiral gear key.
23	2	Rear axle gear case pinion shaft bearings.
24	2	Rear axle gear case pinion shaft bearing washers.
25	1	Rear axle gear case pinion shaft lock nut, R.H.
26	1	Rear axle gear case pinion shaft lock nut lock washer.
27	1	Rear axle gear case pinion shaft lock nut, L.H.
	1	Rear axle gear case pinion shaft lock nut lock washer.
28	1	Rear axle gear case bearing retainer and dust cap (upper).
29	1	Rear axle gear case bearing retainer and dust cap (lower).
30	2	Rear axle gear case bearing retainer locks.
31	1	Rear axle gear case bevel driving gear.
32	1	Rear axle differential case (two halves).
	2	Rear axle differential case bushings.
	4	Rear axle differential case bushing dowels.
33	4	Rear axle differential case bolts.
	8	Rear axle differential case bolt nuts.
	4	Rear axle differential case bolt washers.
34	2	Rear axle differential case gears.
	2	Rear axle differential case thrust collars.
35	1	Rear axle differential case bearing (thrust).
	1	Rear axle differential case bearing.
36	2	Rear axle differential case locks nuts.
37	2	Rear axle differential case lock nut keys (No. 9 Woodruff).
38	4	Rear axle differential gear pinions.
39	4	Rear axle differential gear pinion thrust collars.
40	1	Rear axle differential gear spider.
41	1	Rear axle bracket, R.H.
	1	Rear axle bracket, L.H.
42	2	Rear axle bracket radius rod balls.
	2	Rear axle bracket radius rod ball washers.
43	2	Rear axle bracket radius rod ball nuts.
44	16	Rear axle bracket bolts.
	8	Rear axle bracket bolt lock washers.
45	2	Rear axle spindles.
46	2	Rear axle spindle nuts.
47	2	Rear axle spindle nut washers.
48	2	Rear axle internal brake rings.
49	6	Rear axle internal brake ring shoes.
	72	Rear axle internal brake ring shoe rivets.
	2	Rear axle internal brake ring binder screws.
	6	Rear axle internal brake ring binder screw lock washers.
50	2	Rear axle internal brake ring binder screws.
51	2	Rear axle internal brake ring binder screw jam nuts.
	4	Rear axle internal brake ring adjusting screws.
	4	Rear axle internal brake ring adjusting binder screws.
	4	Rear axle internal brake ring adjusting binder screw nuts.
52	2	Rear axle internal brake ring springs.
53	2	Rear axle internal brake ring studs.

NOTE.—Differential cases are machined together: cannot furnish one half only.

Ref. No.	Number Per Car	Name
54	2	Rear axle internal brake ring stud nuts.
55	4	Rear axle internal brake ring stud bushings.
	2	Rear axle internal brake ring stud bushings.
	2	Rear axle internal brake ring stud washers.
	2	Rear axle internal brake ring stud washers.
56	2	Rear axle foot brake drums.
57	2	Rear axle foot brake bands.
58	2	Rear axle foot brake band shoes (110- × ¼- × 3½-in.).
	24	Rear axle foot brake band shoe rivets.
59	1	Rear axle foot brake band end, R.H.
	1	Rear axle foot brake band end, L.H.
60	2	Rear axle foot brake band ends.
61	2	Rear axle foot brake band end pins.
62	1	Rear axle foot brake band lever, R.H.
	2	Rear axle foot brake band lever bushings.
	1	Rear axle foot brake band lever, L.H.
	2	Rear axle foot brake band lever bushings.
63	2	Rear axle foot brake band adjusting screws.
64	2	Rear axle foot brake band adjusting screw pins.
	4	Rear axle foot brake band adjusting screw pin nuts.
65	2	Rear axle foot brake band adjusting screw collars.
66	2	Rear axle foot brake band adjusting screw washers.
67	2	Rear axle foot brake band adjusting screw springs.
68	2	Rear axle foot brake band adjusting screw jam nuts.
69	2	Rear axle foot brake band adjusting screw nuts.
70	2	Rear axle foot brake band adjusting screw stud guides.
	2	Rear axle foot brake band adjusting screw stud guide nuts.
71	2	Rear axle foot brake band adjusting screw stud guide washers.
72	2	Rear axle foot brake band studs.
	2	Rear axle foot brake band stud nuts.
	2	Rear axle foot brake band stud nut washers.
73		Rear axle foot brake band stud washers.
74	2	Rear axle foot brake band stud adjusting screws.
75	2	Rear axle foot brake band stud adjusting set screws.
	2	Rear axle foot brake band stud adjusting set screw jam nuts.
	4	Rear axle foot brake band stud dowels.
76	2	Rear axle foot brake band stud guides.
77	1	Rear axle brake wing, R.H.
	1	Rear axle brake wing, L.H.
78	2	Rear axle brake wing bushings.
79	2	Rear axle brake wing levers.
80	2	Rear axle brake wing ridge pins.
81	2	Rear axle brake wing ridge pin nuts.
82	1	Rear axle truss rod, R.H.
83	1	Rear axle truss rod, L.H.
84	1	Rear axle truss rod turnbuckle.
85	1	Rear axle truss rod turnbuckle jam nut, R.H.
86	1	Rear axle truss rod turnbuckle jam nut, L.H.
87	2	Rear axle truss rod pins.
88	2	Rear axle wheels (36- × 7-in. single).
89	1	Rear axle wheel hub, R.H.
	1	Rear axle wheel hub, L.H.
90	2	Rear axle wheel hub bushings.
91	1	Rear axle wheel hub lock nut, R.H.
	1	Rear axle wheel hub lock nut, L.H.
92	2	Rear axle wheel hub lock nut washers.
93	2	Rear axle wheel hub flanges.
94	12	Rear axle wheel hub and drum bolts.
	12	Rear axle wheel hub and drum bolt washers (1/16 in. thick).
95	12	Rear axle wheel hub and drum bolt nuts.
96	2	Rear axle wheel hub and dust caps.
97	12	Rear axle wheel hub bolts.
98	12	Rear axle wheel hub bolt nuts.
99	2	Rear axle bearings.
100	2	Rear axle bearing retainers.
101	2	Rear axle bearing retainer felt washers.
	2	Rear axle bearing retainer washers.
102	4	Rear axle bearing retainer hook bolts.
	4	Rear axle bearing retainer hook bolt nuts.
	4	Rear axle bearing retainer hook bolt washers (1/16 in. thick).
103	1	Roller bearing lock screw jam nut.
104	1	Roller bearing ring lock screw.
105	1	Roller bearing ring.
106	1	Roller bearing washer.

The pinion shaft *J* is mounted on two taper roller bearings, which can be adjusted by the two nuts *L*. The pinion shaft and

bearings are all mounted in a separate case *K*, attached to the front cover. The front end of the pinion shaft is splined, upon which a bevel pinion is pressed and held in place by a nut and washer. This bevel pinion meshes with the bevel gear on the jackshaft which forms a unit with the small spur gear *M*, which in turn meshes with the differential spur ring gear *D*.

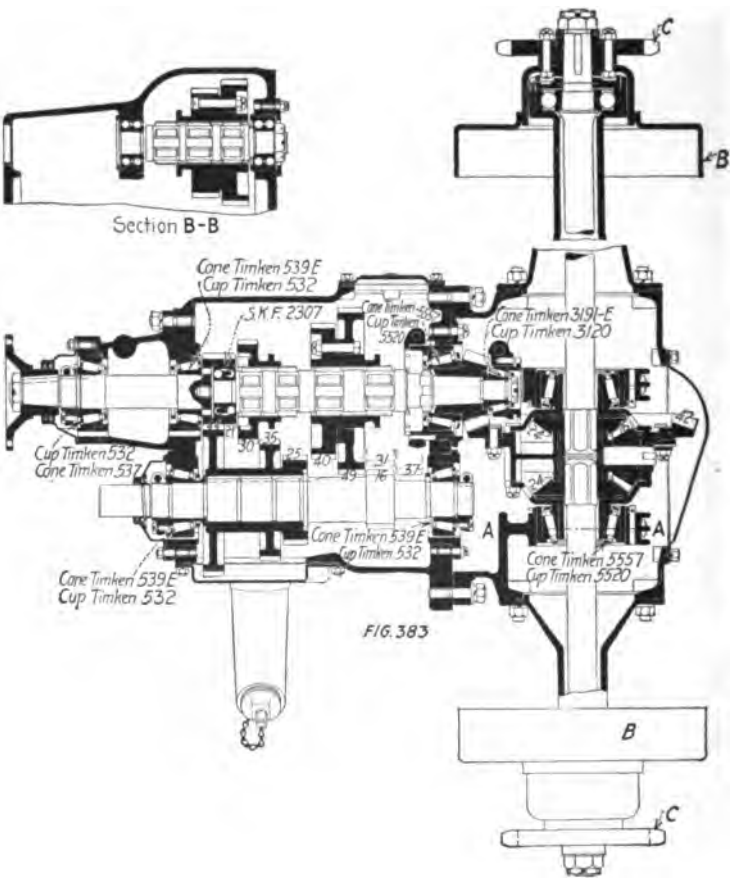


FIG. 383.—Cross-section through transmission and jackshaft of Mack chain driven truck.

By removing plug *P*, the adjustment of the bevel gears, when the cover assembly is in the axle housing, can be examined. The spur gears are located on fixed centers of the casting, into which adjustable bearings are mounted. The whole front cover,

with the exception of the bridge *F* and the pinion shaft case *K*, is a single casting; this bridge is used as a clamp to hold the bearing adjusters and retainers in position. By employing spur gears between the jackshaft and the differential ring gear, there is no side thrust against the ring gear. The axle is of the full-floating type. The wheel is mounted upon the tube which carries the wheel bearings. This tube is pressed into the axle housing under heavy pressure and then held in place by set screws.

In the event of accident, this tube will sustain the damage rather than the axle housing, and it can be replaced at comparatively low cost. In the heavy duty trucks the axle casings are of square section, and the tubes are pressed all the way through the casing into a web, close to the differential gears, thus reinforcing the casing.

Examples of Chain Final Drives.—Figure 383 is a sectional view through the transmission and jackshaft of the Mack chain-driven truck. This type of drive is used by the International Motor Co. on all its heavier models. As may be noted from the drawing, the transmission case is bolted to the jackshaft case as is shown more clearly in Fig. 384, which is a side view of the assembly, *S* being a section through the jackshaft. Figure 385 shows a front view of the transmission (see also Fig 5).

The front end of the transmission case is suspended from a frame cross member by two ¾-in. steel studs (see Fig. 5). The rear is supported by two brackets, riveted to the frame side members, one at each end of the jackshaft housing. In this transmission the upper housing is a special aluminum alloy casting and carries the entire transmission mechanism, the bearing being capped to the case from below. The lower case is of pressed steel and contains the transmission lubricant. Most of the transmission bearings are mounted in separate cages which fit into accurately machined bores of the aluminum case. The driving bevel pinion of the jackshaft has 13 teeth, 3 pitch and a 1½-in. face width, the driven bevel gear has 42 teeth. The differential bevel gears have 24 teeth, 4 pitch, 1⅜-in. face; the three differential pinions have 16 teeth each. The live jackshafts are made of nickel-alloy steel and are 1¾ in. in diameter. The entire transmission and the differential in the jackshaft housing are supported on Timken roller bearings, with the exception of the self-aligning S.K.F. pilot bearing in the pinion gear and the ball bearings at the outer ends of the shafts. The sliding

gears and the counter shaft gears have a 5-6 pitch and a face width of 1½ in The transmission sliding gear shaft is 2 in. in diameter below the splines, and the counter shaft 2⅛ in. Section B-B shows the reverse gears.

The service brake drum hubs are keyed to the outer tapered ends of the shafts and the driving sprockets *C* are bolted to these hubs. The service brakes on drums *B* are of the external contracting type, 5 in. in diameter and 3 in. wide. The emergency

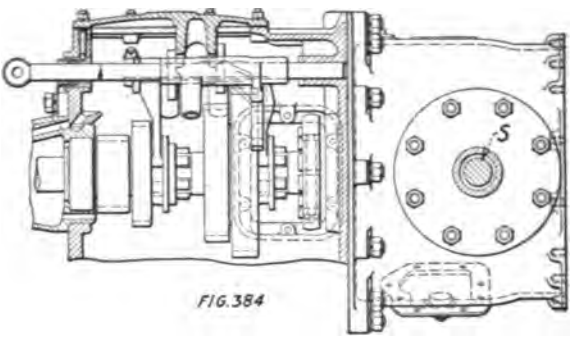


FIG. 384

Section A-A

3½-Ton Sprockets	5, 6½ & 7½-Ton Sprockets
9AC21 - 13 Teeth	9AC23 - 12 Teeth
9AC218 - 14 "	9AC219 - 13 "
9AC210 - 15 "	9AC220 - 14 "
9AC256 - 16 "	9AC28 - 15 "
9AC22 - 18 "	9AC29 - 18 "
9AC27 - 21 "	

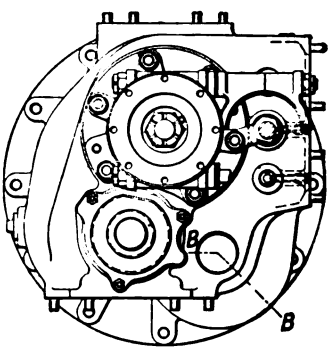


FIG. 385

FIGS. 384 and 385.—Mack chain-driven truck (jackshaft and transmission).

brakes are located in the rear wheels, (see Fig. 386), and are 20 in. in diameter and 3½ in. wide.

The wheels are of the full-floating type and the drive is through a radius rod *E* which also performs the functions of a thrust rod. The rear end of the rod is riveted to the brake supports by rivets *R*; the front end is provided with a ball and socket joint. The large sprockets *D* are integral with the brake drums of the

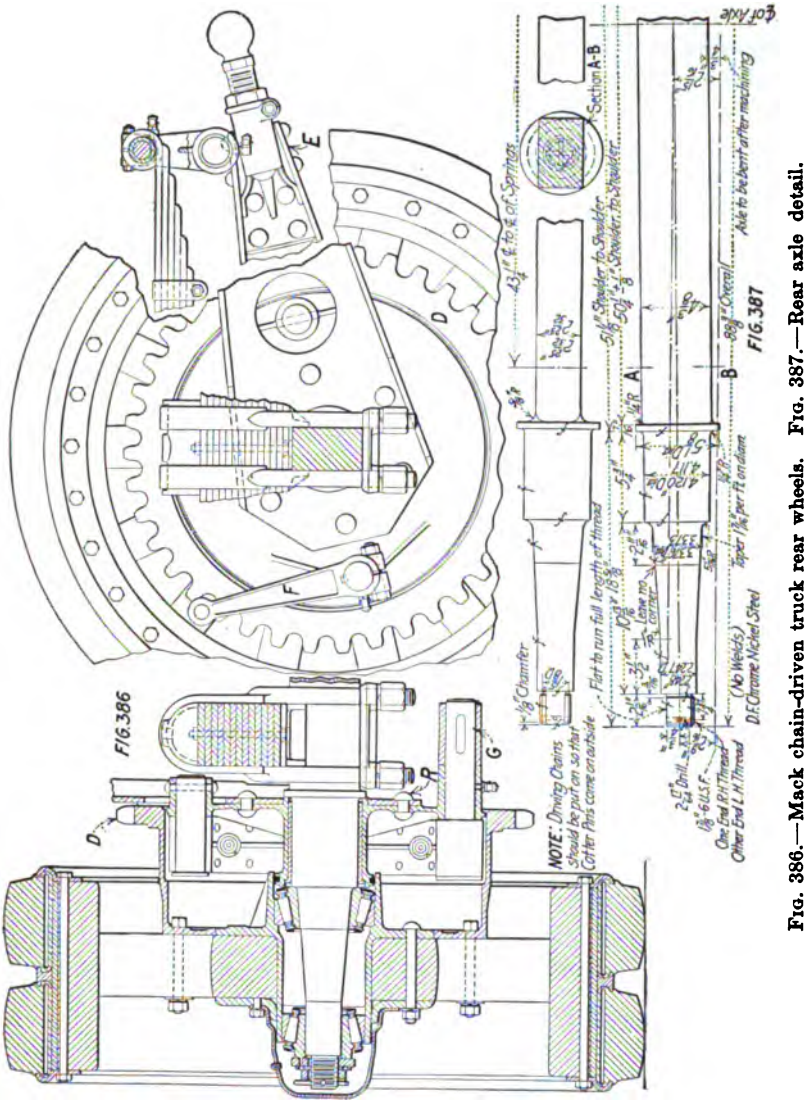


Fig. 386.—Mack chain-driven truck rear wheels. Fig. 387.—Rear axle detail.

rear wheels; the brake is operated by lever F attached to shaft G. In chain driven axles the rear axle is simply a load carrying member or dead axle. A detail of this axle as used on the 5-ton model is given in Fig. 387, showing this axle to be of rectangular section, $4\frac{3}{8}$ in. deep and $2\frac{3}{4}$ in. wide. The axle of the $3\frac{1}{2}$-ton model is $2\frac{1}{4}$ in. wide by $3\frac{5}{8}$ in. high, while in the $7\frac{1}{2}$-ton axle it is $2\frac{3}{4} \times 4\frac{3}{4}$ in. It is dropped forged, or chrome-nickel steel, heat treated. The axle is slightly arched, being $\frac{3}{8}$ in. higher in the center than at the spindle.

CHAPTER IX

LOADS AND STRESSES, REAR AXLE

Loads on Wheel Bearings and Stresses in Rear Axles (More especially for passenger cars).—(See pages 328 to 335 for bearing load and shaft stresses in truck rear axles.) When a motor vehicle is standing still, the static load on each rear wheel is one-half the weight resting on the rear axle. When it is travelling at a certain speed, the load on the wheels varies when they surmount obstructions on the road and when sinking into ruts; it also varies considerably when the car is turning a corner or when travelling around a curve. In such event the centrifugal force will tend to move or skid the wheels sidewise, to raise the inner wheel off the ground and throw additional load on the outer wheel.

In Vol. I, Chap. XII, it was shown that the centrifugal force $C = \frac{WV^2}{gR}$. In the present case W is the load or weight resting on the rear axle; V the speed or velocity in feet per second; g, the acceleration due to gravity (32.2); and R, the radius in feet of the curve described by the center of gravity of the weight carried by the rear axle.

For instance, if the car travels around a circle having a radius of 30 ft., at a speed of 15 m.p.h., $V = \frac{15 \times 5{,}280}{60 \times 60} = 22$ ft. per second, (60×60 = number of seconds per hour). If the load on the rear axle is, for instance, 2,400 lb., the centrifugal force $C = \frac{2{,}400 \times 22^2}{32.2 \times 30} = 1{,}202$ lb., and this is the force which will tend to skid the rear of the car sidewise. This force will be resisted by the frictional force F at the road surface, and this force F is therefore acting in a direction opposite to that of the centrifugal force C and is always equal to it. From the above equation it is seen that the centrifugal force C increases as the square of the velocity V (*i.e.*, if V is increased to two times the original speed, C is increased to 2×2, or four times its former

value), while it changes inversely with the radius (R it decreases in direct proportion with an increase in R); if the radius were doubled, C would be halved and *vice versa*.

When the centrifugal force C reaches a certain magnitude, skidding takes place, and this occurs when C is equal to the force of friction between tires and ground. This equals (on the rear) to the total load W on the rear wheels multiplied by the coefficient of friction c. Hence, we can write, when skidding occurs; $\frac{WV^2}{gR} = cW$, from which $V^2 = \frac{cWgR}{W} = cgR$, and $V = \sqrt{cgR}$; and $R = \frac{V^2}{gc}$. In the case before cited, the velocity was 22 ft. per second and the coefficient of friction between tires and road surface .6, therefore the maximum radius R at which skidding occurs (when traveling at 22 ft. per second) $= \frac{V^2}{gc} = \frac{22^2}{32.2 \times .6} =$ 25 ft. (If this radius is substituted for the 30 ft. mentioned, in the formula before used, C would = 1,440 lb.) When the radius is given, the maximum speed at which skidding takes place can be found from the equation given above; $V = \sqrt{cgR}$. As an example, if the turning radius R is 30 ft., $V = \sqrt{.6 \times 32.2 \times 30} =$ 24 ft. per second approximately, (or $\frac{24 \times 60 \times 60}{5{,}280} = 16.3$ miles per hour). From these formulas it is also possible to find the speed at which the car will begin to turn over, if the car is prevented from skidding or if the coefficient of friction is high enough. This will take place when all the load rests on the outer wheel, and the inner wheel is lifted off the ground, *i.e.*, if $P_1 = O$, as will be explained later.

In practice, when skidding takes place, the centrifugal force is usually smaller, because the road is slippery and therefore the coefficient of friction is much less, but as we are trying to find the maximum bearing pressures, we must consider a dry road surface. If the coefficient of friction between tires and road is .6, then at the time of skidding, $C = .6 \times W = .6 \times 2{,}400 = 1{,}440$ lb.

If the centrifugal force is located at a height h from the ground, (see Fig. 388), if the total pressure between the two rear tires and the ground be designated by P (P_o is total pressure of outer wheel and P_i of inner wheel, therefore $P = P_o + P_i$), and the tread

by t, then by taking moments about the point where the outer wheel touches the ground, we have positive moments $W \times \frac{t}{2}$, and this equals the negative moments $C \times h + P_i \times t$; hence $W \times \frac{t}{2} = Ch + P_i t$, from which

$$P_i = \frac{W \times \frac{t}{2} - C \times h}{t} = \frac{W \times \frac{t}{2}}{t} - \frac{Ch}{t} = \frac{W}{2} - \frac{Ch}{t}.$$

We thus see that the pressure P_i between inner wheel and ground is smaller by $\frac{Ch}{t}$ than one-half the weight resting on the rear axle. Assume C to be so large as to cause skidding, then $C =$

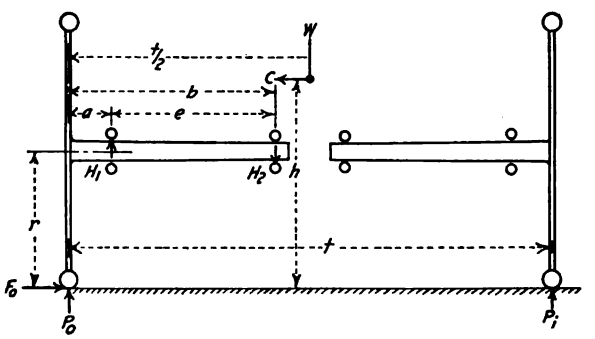


FIG. 388.—Semi-floating axle.

$F_o + F_i$; and this must also equal to $cW = .6 \times 2{,}400 = 1{,}440$ lb. Taking moments about the points of contact of the inner wheel with the ground, in the same manner as before, we have

$$P_o \times t = W \times \frac{t}{2} + Ch,$$

from which

$$P_o = \frac{W \times \frac{t}{2} + Ch}{t} = \frac{W}{2} + \frac{Ch}{t};$$

thus the radial load on the outer wheel is increased when going around a curve by a considerable amount; substituting figures

for the letters, and assuming h to be 24 in. from the ground, and the tread t to be 56 in.,

$$P_o = \frac{2,400}{2} + \frac{1,440 \times 24}{56} = 1,200 + 617 = 1,817 \text{ lb.}$$

$\frac{Ch}{t} = 617$, therefore on the inner wheel, the actual radial load is $P_i = 1,200 - 617 = 583$ lb. (See formula for P_i given before.)

In addition to the radial load, there is a thrust load F on the two wheels arising from the tendency to skid the rear wheels, this thrust load is equal to the centrifugal force C, and it is divided between the two rear wheels in proportion to their radial load. $C = F$, and $F = F_o + F_i$ that is to say, F equals the side thrust on the outer wheel, plus the side thrust on the inner wheel, (but F_o is not equal to F_i). Since the coefficient of friction between rubber tires and the average road surface is about .6 therefore on the outer wheel the side thrust $F_o = 1,817 \times .6 = 1,090.2$ lb., and on the inner wheel $F_i = 583 \times .6 = 349.8$ lb.

If was shown under chapter on "Clutches," that the coefficient of friction c equals the horizontal force (in this case C) required to move a certain load, divided by the amount of this load, in this case W, and $W = P$, *i.e.*, the total pressure between the two rear tires and the road. Therefore, when the centrifugal force is as large as to cause the rear of the car to skid, we have coefficient of friction $c = .6 = \frac{C}{P}$, (P being $P_i + P_o$), and for the outer rear wheel $.6 = \frac{F_o}{P_o}$ and $F_o = .6P_o$. As the radial load on the outer tire is greater than on the inner tire, we shall only consider that of the outer wheel for finding the maximum side thrust F_o, at the bottom of (the outer) wheel when the tire skids. Substituting figures in the last equation for P_o found above, we have $F_o = .6 \times 1,817 = 1,090$ lb. Thus we have a total radial load on the rear wheel of 1,817 lb., and a side thrust of 1,090 lb.

Bearing Loads in Full-floating Axle.—The next step is to determine the maximum load on the two wheel bearings H_1 and H_2 under these conditions. First consider the full-floating axle, (see Fig. 389). Taking moments about H_1, we have $P_o \times b = H_2 e$, hence $H_2 = \frac{P_o \times b}{e}$. But in addition to find the maximum load we must consider the effect of the side thrust F_o on the bearings, since this thrust tends to cause the wheel to assume the

position shown by the dotted line Y. Taking moments about points H_1, and considering both the radial and thrust loads, we have $H_2 e = P_o b + F_o r$, from which $H_2 = \frac{P_o b + F_o r}{e}$. If the distance e between the bearings is 4.5 in., if a is 1.75, and b, 2.75 in., and if r the radius of the wheel, is 18 in., the radial load or reaction $H_2 = \frac{1{,}817 \times 2.75 + 1{,}090 \times 18}{4.5} =$ 5,468 lb. In addition, bearing H_2, the double row bearing, is taking the thrust, (bearing H_1 is not intended to take any thrust in this case) therefore it also has to withstand the side thrust of 1,029 lb, as found before. Instead of a double row bearing, sometimes a single row bearing or a roller bearing is here used.

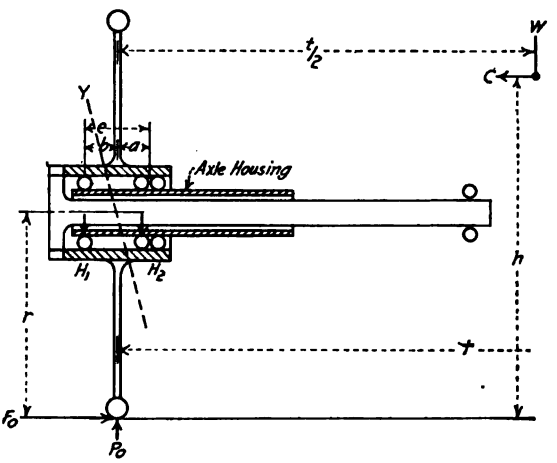


FIG. 389.—Full-floating axle.

To find the resultant bearing load R arising from the radial load and the thrust, (they are at right angles to each other), we have, according to the law of the right angled triangle,

$R = \sqrt{5{,}468^2 + 1{,}090^2} = \sqrt{31{,}085{,}000} = 5{,}570$ lb. approximately.

To find the pressure on bearing H_1, take moments about the right hand bearing; the positive moment is $P_o a$; the negative moments are $H_1 e + F_o r$, from which $H_1 = \frac{P_o a - F_o r}{e}$. Substituting figures,

$$H_1 = \frac{1{,}817 \times 1.75 - 1{,}990 \times 18}{4.5} = -3{,}655 \text{ lb.}$$

It should be noted that the answer is a negative quantity, which implies that there is a negative load on this bearing *i.e.*, the load is not resting on the usual side of the bearing, as will be shown later. Considering the effect of the side thrust, it is evident that this will tend to relieve the bottom of the bearing H_1 of load or pressure and impart it to the top of the bearing. Since the second term of the equation $-F_o r$, is considerably greater than the first term $P_o a$, it is evident, that not only is the bottom of the bearing relieved of all pressure, but a considerable pressure will exist on the top of the bearing, which place usually does not carry any load. Since the single row bearing in this case is not mounted to take any thrust, it need not be considered for bearing H_1.

The bearing pressures found so far are those due to skidding, when traveling around a corner at a speed which induces a centrifugal force. When hitting a curb or a car rail, the coefficient of friction will be considerably higher, in fact the wheel may be stopped from moving sidewise entirely, in which event the strain against the bearings may be considerably higher, the wheel may break or the car may turn over.

We will now consider the bearing loads due to straight forward travel when the brakes are applied. The ordinary radial load P on each rear wheel, may be represented by a vertical or radial force acting in the same direction as shown by the arrow P_o, and this causes the lower side of the bearings to support the shaft (in this case the axle tube). When the brakes are applied a horizontal force will be imparted to the wheel in addition to the radial force. If the wheel is locked, the horizontal force at the tire (and also at the wheel centers) = the coefficient of friction c, multiplied by the load on the wheel which is the pressure P between tire and ground, (when the car is traveling in a straight line). The normal load on the wheel or the pressure P in such event is $\frac{W}{2}$ *i.e.*, one-half of the weight resting on the rear axle. (This is not strictly true when the car is travelling, but it may be assumed for all calculations.) Hence the horizontal force = $c\,\frac{W}{2}$. The resultant force of any two forces acting at right angles to each other, can be found from the well known formula, hypothenuse squared = the sum of the squares of the two forces at right angles to each other.

If P is the ordinary radial load or force on the wheel $\left(=\frac{W}{2}\right)$ P_h the horizontal radial force when the wheel is locked, and P_r the resultant radial force, then

$$P_r = \sqrt{P^2 + P_h^2} = \sqrt{\left(\frac{W}{2}\right)^2 + \left(c \times \frac{W}{2}\right)^2}.$$

Substituting figures

$$\sqrt{\left(\frac{2{,}400}{2}\right)^2 + \left(.6 \times \frac{2{,}400}{2}\right)^2} = \sqrt{1{,}440{,}000 + 518{,}000} = 1{,}399 \text{ lb.}$$

In practice, the bearing spacing for pleasure cars is approxi-

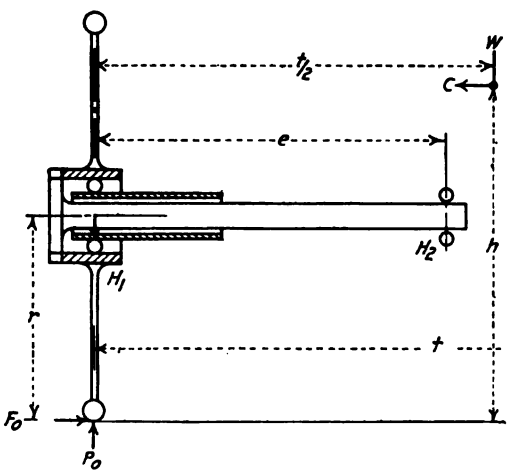


FIG. 390.—Three-quarter floating axle.

mately as the dimensions given in the above example. From data gathered for this type of wheel mounting in practice, it is found that for the inside bearing which also takes the thrust, a bearing whose capacity is from about 70 per cent to 90 per cent of the maximum load, (when skidding) is usually employed, while a bearing with a capacity of from about 55 per cent to 75 per cent of the value found, is required for the outside bearing. If larger bearings are used, it entails unnecessary cost, and since the side thrust which causes the maximum bearing pressures occurs only occasionally, it is as a rule, not necessary to use larger bearings. In the selection of bearings, it should be remembered that the number of revolutions must be considered in addition to the loads, to determine the correct bearing capacities. The class of service of the motor vehicle and the type of bearing also influence

the capacity, hence it is advisable to consult the bearing manufacturers whose bearings it is intended to employ.

Bearing Loads in Three-quarter Floating Axles.—To find the maximum bearing pressures in this type of axle, (see Fig. 390), at first the same method may be pursued as before, and the radial load on the outer wheel, when turning a corner or travelling on a curve, may be determined; we thus have $P_o = \frac{W}{2} + \frac{Ch}{t} = 1{,}817$ lb., when W, C, h and t have the same values as before. Likewise $F_o = .6 \times 1{,}817 = 1{,}090$ lb.

Taking moments about the right hand bearing and disregarding the side thrust at first, we have (positive moment) $P_o e$, which is equal to (the negative moment) $H_1 e$; in other words, the bearing load is equal to the load resting on the tire. When the thrust F_o is considered, we have, by taking moments about H_2, the negative moment $F_o r$ in addition. Hence $P_o e = H_1 e + F_o r$, and $H_1 = \frac{P_o e - F_o r}{e} = P_o - \frac{F_o r}{e}$. If the distance between the bearings is 25 in. and the other values as before given by substituting figures, $H_1 = 1{,}817 - \frac{1{,}090 \times 18}{25} = 1{,}032$ lb.

When taking moments about H_1 it is found that $H_2 e = F_o r$, hence $H_2 = \frac{F_o r}{e} = \frac{1{,}090 \times 18}{25} = 785$ lb., and this load will be supported by the top of the bearing when the car skids, or when the wheel is subjected to a large thrust load, while under certain conditions of straight travel, when there are only radial loads on the wheel, there is no load on bearing H_2. Thus the maximum pressure on the inner bearing is 785 lb. when the car skids sidewise.

On straight forward travel, the radial load on the wheels is the same as in the full floating axle, *i.e.*, if the static load on the wheel $P = \frac{W}{2}$, and if P_h is the horizontal load when the wheel is locked, then the resultant radial load or force

$$P_r = \sqrt{P^2 + P_h^2} = \sqrt{\left(\frac{W}{2}\right)^2 + \left(c\,\frac{W}{2}\right)^2} = 1{,}399 \text{ lb.}$$

(see example worked out for full-floating axle).

In the three-quarter floating axle, the outer bearing is in the same plane as the radial load on the wheels, *i.e.*, it is in line with

the wheels, hence the outer bearing carries the entire load under these conditions and this load, 1,399 lb., is greater than when the car skids, which, as was shown, is 1,032 lb. According to the cars on the market, in the selection of this bearing a factor of safety of from about 2 to 4 is used, (depending on the class of service, etc.), in other words, a bearing should be selected which, if we assume a factor of safety of 3½, would have a capacity of 3½ × 1,399 lb. = 4,896 lb. A high factor of safety is here necessary, because the ordinary load $\left(\frac{W}{2} = 1{,}200 \text{ lb.}\right)$ is continuous, and when the wheel sinks into ruts or hits obstructions, the radial load is considerably higher. The reason for the reduced load when skidding is that the two bearings are far apart. Manufacturers should submit their designs to bearing manufacturers, for their recommendations as to bearing capacities. With this construction (three-quarter floating) the live axle must be positively and firmly attached to the wheel hub, else, when the car skids, the shaft will not transmit a portion of this load to the differential bearing but the wheel bearing will be strained excessively and cramped sidewise. For this reason, the fit of the live axle in the differential must not be more than necessary for a good running fit, and the live axle itself must be of sufficient strength that it deflect very little when skidding takes place. The practical designs given in this book should be studied by students.

Bearing Loads in Semi-floating Axles.—(See Fig. 388.) In this axle, P_o and F_o are the same as before, *i.e.*, 1,817 lb. and 1,090 lb. respectively. When the wheels skid the bearing pressures are found as follows: (Consider the outside wheel only, as this wheel carries the maximum load.) Taking moments about H_2, we have $P_o b = H_1 e + F_o r$, from which $H_1 = \frac{P_o b - F_o r}{e}$. If the distance e between the two bearings is 22 in., the distance a between the outer bearing and the wheels 3 in., then b is 25 in. and

$$H_1 = \frac{1{,}817 \times 25 - 1{,}090 \times 18}{22} = 1{,}173 \text{ lb.}$$

By taking moments about H_1, we find $P_o a = H_2 e + F_o r$, from which

$$H_2 = \frac{P_o a - F_o r}{e} = \frac{1{,}817 \times 3 - 1{,}090 \times 18}{22} = -644 \text{ lb.}$$

The result is minus, as the bearing will carry the load on the top, while under normal running, it will carry it at the bottom.

Under straight forward travel, the resultant radial load P_r on the wheel will be as before, *i.e.*, 1,399 lb. Taking moments about bearing H_2 with this resultant radial load, we have $P_r b = H_1 \times e$, from which

$$H_1 = \frac{P_r b}{e} = \frac{1{,}399 \times 25}{22} = 1{,}589 \text{ lb.},$$

and taking moments about H_1, we find $P_r a = H_2 e$, from which

$$H_2 = \frac{P_r a}{e} = \frac{1{,}399 \times 3}{22} = 190.5 \text{ lb.}$$

In the semi-floating construction, the live axle is subjected to combined bending and torsion stresses, as will be shown later, see page 330. Note that in this type, when running straight ahead the load on the bearings is greater than when skidding. From a number of models on the market it is found that a factor of safety of from about 2 to 3 is required in the outer bearing for satisfactory performance. The capacity varies according to design of vehicle, service, type of bearing, etc. When considering the inner or differential bearing, the load arising from the final drive or the bevel gears must also be considered and added to the load induced through the wheel. For shaft stresses and other examples of figuring bearing loads, see pages 329 to 335.

Bearing Loads and Rear Axle Shaft Stresses (more especially for trucks).—To find the bearing loads and shaft stresses in semi-floating rear axles, three distinct conditions must be considered: First, the maximum torque + the normal radial load on the wheel; second, the wheel locked and skidding forward (when the brakes are applied) and third, the wheel skidding sidewise while the truck is running.

For example, a certain truck axle has a maximum total reduction of 49.61; the maximum hp. is 50 at 1,000 r.p.m., and and the load on each rear wheel 9,000 lb. The torque in the rear axle shaft, when allowing a transmission efficiency of 85 per cent and running in low gear is,

$$T = \frac{\text{hp.} \times 63{,}025 \times 49.61 \times .85}{1{,}000} = 132{,}850 \text{ lb.-inches}$$

or one-half this amount, 66,425 lb.-inches, in each half of the rear axle shaft.

On the other hand, if a transmission brake is employed, or if the engine is speeded up and the clutch thrown in suddenly, causing the rear wheels to slip on the ground, the maximum force at the periphery of the wheel is .6 × 9,000 = 5,400 lb. (.6 being the coefficient of friction between tire and road surface) and if the wheel diameter is 36 in., the torque in each half of the shaft is 18 × 5,400 = 97,200 lb.-inches. However, we will not consider these conditions, for in our example no propeller shaft brake is employed, and steel can stand a large momentary, occasional overload, without serious damage.

The shearing stress in the shaft (torsional stresses are shearing stresses) can be found from the well known formula $T = S_s Z_p$, where T is the torque, S_s the shearing stress in pounds per square inch and Z_p the polar section modulus, from which $S_s = \frac{T}{Z_p}$ (see page 119).

The diameter at the weakest point of the axle (close to the inner bearing) is $2\frac{7}{16}$ in. = 2.4375 in. (semi-floating axles are made tapered like that shown in Fig. 327). The polar section modulus at this point is $Z_p = .1963D^3 = .1963 \times 2.4375^3 = 2.84$. Hence, shearing stress $S_s = \frac{T}{Z_p} = \frac{66{,}425}{2.84} = 23{,}390$ lb. per square inch, (due to torsion). At the outer bearing the shaft is 4 in. in diameter, therefore $Z_p = .1963 \times 4^3 = 12.56$, and $S_s = \frac{66{,}425}{12.56} = 5{,}280$ lb. per square inch approximately.

We will now consider the bearing loads and shaft stresses due to the bending moments. Under ordinary running conditions, the radial load on each wheel is 9,000 lb.; the distance e between the bearing centers is 25 in. (see Fig. 388) and between the outer bearing and the center of the wheel $a = 6$ in. (distance $a + e = 31$ in.). If H_1 is the reaction or the radial load on the outer bearing, and H_2 that of the inner or the differential bearing, and if the radial load on the wheel is designated by P, then taking moments about the outer bearing, we have $6P = 25H_2$, and

$$H_2 = \frac{6P}{25} = \frac{6 \times 9{,}000}{25} = 2{,}160 \text{ lb.}$$

Taking moments about the inner bearing, $31P = 25H_1$, and

$$H_1 = \frac{31P}{25} = \frac{31 \times 9{,}000}{25} = 11{,}160 \text{ lb.}$$

The bending moment in the shaft near, or one inch from, the

inner bearing, is 2,160 lb.-inches, increasing toward the outer bearing, where it is $2,160 \times 25 = 54,000$ lb.-inches.

The tensile and compressive stress S of a shaft is found from the formula $S = \frac{B}{Z}$, where B is the bending moment and Z the section modulus. (See pages 44 to 46 for section modulus.) The section modulus, (not polar section modulus) of a round shaft is $.098 \times \text{diameter}^3$. The shaft diameter near the outer bearing is 4 in., therefore, here, the section modulus $Z = .098 \times 4^3 = 6.28$. Hence, at this point $S = \frac{54,000}{6.28} =$ 8,600 lb. per square inch approximately. Near (1-in. from) the inner bearing $Z = .098 \times 2.4375^3 = 1.418$, and $S = \frac{2,160}{1.418} =$ 1,523 lb. per square inch.

To find the stress in a shaft subjected to torsion and bending stresses, we may make use of an equivalent twisting moment T_e, which would create the same stress in the shaft as that due to the combined twisting moment T, and the bending moment B. The equivalent twisting moment $T_e = \sqrt{B^2 + T^2}$. Near the inner bearing $T_e = \sqrt{2,160^2 + 66,425^2} = 66,500$ lb.-inches approximately, almost the same as that found for the twisting moment only. At the outer bearing $T_e = \sqrt{54,000^2 + 66,425^2} =$ 85,000 lb.-inches approximately. The stress in the shaft at the outer bearing is therefore $S_s = \frac{T_e}{Z_p} = \frac{85,000}{12.56} = 6,570$ lb. per square inch approximately. While at the inner bearing $S_s = \frac{66,500}{2.84} =$ 23,400 lb. per square inch approximately, showing that the stress in the shaft is much greater near the inner bearing since its area is here much smaller. (T_e must be divided by the *polar* section modulus Z_p to find the shearing stress S_s.)

The material used for this shaft is S.A.E. 2, 340 steel, which, if heat treated to a brinnell hardness of 335, or a scleroscope hardness of 51, has a tensile strength of 175,000 lb. per square inch, and elastic limit of 150,000 lb. per square inch, according to the S.A.E. data sheets. The shearing strength of steel is approximately 85 per cent of its tensile strength, or in our case, $175,000 \times .85 = 149,000$ lb. per square inch, while the transverse elastic limit or the elastic limit in shear, is approximately 35 per cent of its ultimate shearing strength, and this should be considered when

finding the real factor of safety. Therefore, the elastic limit in shear = 149,000 × .35 = 52,150 lb. per square inch approximately. (There seem very few data available as to the elastic limit in shear of various steels.) Hence, at the weakest portion of the shaft, the factor of safety is $\frac{52,150}{23,400} = 2.23$ approximately under normal conditions, when running in low gear. Frequently, the ultimate strength of the material is used, but the author advises that the elastic limit be considered, as it is a much safer guide.

Next, we will investigate the bearing loads and shaft stresses when the brakes are applied and the wheels locked and sliding forward. In this case there is in addition to the static or normal load of 9,000 lb., a horizontal force at the periphery of the wheel equal to the vertical or normal load resting on it, multiplied by the coefficient of friction, as was mentioned before; thus .6 × 9,000 = 5,400 lb. The two forces, the vertical or static and the horizontal act at right angles to each other. The resultant radial load R on the wheel due to these two forces is,

$$R = \sqrt{9,000^2 + 5,400^2} = \sqrt{110,100,000} = 10,500 \text{ lb.}$$

If we call the reactions (bearing pressures) in the outer and inner bearings, H_1 and H_2, respectively, by taking moments about the outer bearing, we have $6R = 25H_2$, from which

$$H_2 = \frac{6R}{25} = \frac{6 \times 10,500}{25} = 2,520 \text{ lb.}$$

Taking moments about the inner bearing, we have $31R = 25H_1$, hence, $H_1 = \frac{31 \times 10,500}{25} = 13,000$ lb. The bending moment near the inner bearing is 2,520 lb.-inches, and near the outer bearing 25 × 2,520 = 63,000 (or 6 × 10,500 = 63,000 lb.-inches). Then, since $S = \frac{B}{Z}$, at the inner bearing, $S = \frac{2,520}{1.418} =$ 1,770 lb., and at the outer bearing $S = \frac{63,000}{6.28} = 10,000$ lb. per square inch approximately (Z, the section modulus, at the inner end of the shaft being 1.418, and at the outer end 6.28 as found before). It is seen therefore, that under these conditions, (wheels locked and sliding forward) the stresses in the shaft are very low, especially in view of the fact that the shaft is subjected to bending stresses only, the material is in tension and compression, and its elastic limit 150,000 lb. per square inch.

We will now consider the bearing loads and shaft stresses resulting from skidding sidewise on a dry road, when turning a corner, at a certain speed. Under these circumstances, the centrifugal force C, which causes the rear of the truck to skid, will equal the coefficient of friction (.6) multiplied by the total weight W carried by the two rear wheels. In our case $C = .6 \times 18{,}000 = 10{,}800$ lb. (See page 321.)

Assuming that the center of gravity of the entire load on the rear wheels is located 45 in. from the ground and calling the distance h, that the total pressure between the two rear tires and the ground is designated by P, that P_o is the pressure on the outer wheel, P_i that on the inner wheel (when rounding a curve), that the tread t, is taken as 56 in., and that the radius r of this wheel is 18 in., then, by taking moments about the point where the inner wheel touches the ground, we have (see Fig. 388)

$$P_o \times t = W \times \frac{t}{2} + C \times h,$$

from which

$$P_o = \frac{W \times \frac{t}{2} + Ch}{t} =$$

$$\frac{W}{2} + \frac{Ch}{t} = \frac{18{,}000}{2} + \frac{10{,}800 \times 45}{56} = 9{,}000 + 8{,}700 = 17{,}700 \text{ lb.}$$

(The load on the inner wheel will be $9{,}000 - 8{,}700 = 300$ lb., see page 321.)

In addition to the radial load, there is a thrust load F on the two wheels, arising from the tendency to skid; this thrust load is equal to the centrifugal force C, and is divided between the two rear wheels in proportion to their radial load. Hence, the maximum thrust load on the outer rear wheel between tire and ground (when the car skids on a dry road) $F_o = .6 \times P_o = .6 \times 17{,}700 = 10{,}600$ lb. The total radial load on the outer wheel is 17,700 lb., and the side thrust at the bottom of the wheel, 10,600 lb.

Taking moments about the inner bearing, we have

$$31P_o = 25 \times H_1 + rF_o,$$

from which

$$H_1 = \frac{31P_o - rF_o}{25} = \frac{31 \times 17{,}700 - 18 \times 10{,}600}{25} = 14{,}300 \text{ lb.},$$

and taking moments about the outer bearing

$$6P_o = 25H_2 + rF_o,$$

hence

$$H_2 = \frac{6P_o - rF_o}{25} = \frac{6 \times 17{,}700 - 18 \times 10{,}600}{25} = -3{,}384 \text{ lb.}$$

The result is a minus quantity, as the bearing will carry the load on the top, while under normal running it will carry it at the bottom. (It should be remembered that the outer bearing also has to carry a thrust load of 10,600 lb.).

Knowing the maximum bearing pressures (the reactions) we can easily find the bending moments and the stresses in the shaft, as in the last example. The bending moment B near (one inch from) the inner bearing is 3,384 lb.-inches, and near the outer bearing, $25 \times 3{,}384 = 84{,}600$ lb.-inches approximately.

In addition to the bending moment due to skidding, the twisting moment in the shaft due to the drive must be considered. Evidently the truck will have to be in the high gear to travel at the requisite speed to produce skidding on a dry road, hence the torque will be much lower, but even if it were running in the low gear, and the torque be 66,425 lb.-inches, the maximum stress in the shaft would be barely higher than under *straight forward* travel, when in the low gear. In practice a truck does not, as a rule, skid sidewise on a dry road surface, but we consider such a condition in order to find the maximum bearing loads.

If the low gear torque is added, the equivalent twisting moment near the inner bearing is $T_e = \sqrt{B^2 + T^2} = \sqrt{3{,}384^2 + 66{,}425^2}$ = practically the same as found before, *viz.*, 66,500 lb.-inches, and $S_s = \frac{6{,}6500}{2.84} = 23{,}400$ lb. per square inch.

Near the outer bearing $T_e = \sqrt{84{,}600^2 + 66{,}425^2} = 107{,}000$ lb.-inches approximately, and the stress $S_s = \frac{107{,}000}{12.56} = 8{,}550$ lb. per square inch. When skidding sidewise the torque is considerably lower in practice, for the truck would not skid unless it traveled in one of the higher speed gears, unless the road surface were muddy or slippery, and in such event the coefficient of friction is considerably lower, hence the bending moment would be lower. The maximum stress (in shear) is therefore near the inner bearing, on straight forward travel in the low gear, and amounts to 23,400 lb. per square inch.

Full-floating Axles.—(See Fig. 389.) The stresses in the shaft of a full-floating axle are purely torsional, *i.e.*, shearing stresses, and if the engine power and gear reduction are as in the last example, the maximum torque will be as found before, *viz.*, 66,425 lb.-inches in each axle shaft. If the shaft dimensions are the same as in the previous example, the maximum unit stress (torsional or shearing) in the shaft metal would be as that found before, 23,390 lb. per square inch.

In a full-floating axle, the shaft dimensions are uniform, as a rule, not tapered as in the semi-floating type, since the live axle is relieved of all bending moments by the wheel bearings.

The maximum radial load on the wheels, when the brake is applied and the wheels slide forward, is the resultant radial load R, due to the combined horizontal and vertical forces and is the same as found before, (R = 10,500 lb.). If the distance e between the wheel bearings is 7 in., the inner bearing (H_2) from the wheel center $a = 2\frac{1}{2}$ in., and from the outer bearing (H_1) $b = 4\frac{1}{2}$ in., then, by taking moments about the outer bearing, we find $7H_2 = 4\frac{1}{2} \times 10{,}500$, and $H_2 = \dfrac{4.5 \times 10{,}500}{7}$ = 6,750 lb.; the load on the outer bearing $H_1 = \dfrac{2\frac{1}{2} \times 10{,}500}{7}$ = 3,750 lb. When the wheels skid sidewise, the bearing loads (on the outer wheel) are considerably higher. If the force of friction or the thrust load F_o on the outer wheel due to the centrifugal force that produces the skidding, and the radial load P_o, are as found before, *i.e.*, 17,700 lb. and 10,600 lb. respectively, then by the law of moments, we obtain $7H_2 = 4.5P_o + 18F_o$, hence

$$H_2 = \frac{4.5 \times 17{,}700 + 18 \times 10{,}600}{7} = 38{,}700 \text{ lb.}$$

and

$$7H_1 + 18F_o = 2.5\,P_o,$$

from which

$$H_1 = \frac{2.5 \times 17{,}700 - 18 \times 10{,}600}{7} = -21{,}000 \text{ lb.}$$

approximately showing that on the outer bearing the load is reversed from its normal direction. In addition, there is a thrust load of 10,600 lb. Tabulating the results we find the following:

	Full-floating axle	Semi-floating axle
Torsional stress in shaft, due to maximum torque....	(Diameter of shaft, $2\frac{7}{16}$ in.) 23,390 lb. per square inch	(Diameter at weakest section, $2\frac{7}{16}$ in.) 23,390 lb. per square inch
Maximum stress due to combined maximum torque and bending moment.....	No bending moment	23,400 lb.
Wheels locked and sliding forward; bearing pressure in H_1.	3,375 lb.	13,000 lb.
Wheels locked and sliding forward; bearing pressure in H_2.	6,750 lb.	2,520 lb.
Truck turning a corner, wheels skidding sidewise; bearing pressure in H_1.....	−21,000 lb.	14,300 lb.
Truck turning a corner, wheels skidding sidewise; bearing pressure in H_2.....	38,700 lb.	−3,380 lb.
Maximum thrust load.....	10,600 lb.	10,600 lb.

From this table we can judge that the maximum shaft stresses are practically the same in both designs, but the shaft in the full-floating axle can be made lighter and at less cost, since the stress is uniform in the entire shaft, and the shaft is relieved of all but torsional stresses; furthermore, in the semi-floating axle, there are bending stresses which are continually reversed, therefore a higher factor of safety should be used.

The bearing loads in the full-floating axle are considerably higher, which will impose a much greater bending moment on the axle housing, hence the bearings and the axle housing must be made heavier and this axle will thus be more expensive to manufacture than the semi-floating axle.

CHAPTER X

TORQUE ARMS, THRUST-, RADIUS-, AND TRUSS RODS

Torque Arms.—The function of the torque arm is to take the torque reaction from the drive. When the wheels are revolved forward by the live axle, in passenger cars through the bevel ring gear, the bevel pinion with its shaft, which is supported in bearings forming a part of the axle housing, will tend to rotate the latter in the opposite direction to that of the bevel ring gear. In other words, the twisting moment tends to propel the wheel forward, while the "reaction" produced tends to rotate the axle housing in the opposite direction, and the function of the torque arm is to resist this torque reaction. When the brakes are applied, the wheels tend to rotate the axle housing in a forward direction (this also occurs when the car is driven backwards) the torque arm must therefore resist the reaction of the housing in both directions.

The torque arm is securely attached to the rear axle housing, while its front end is supported by a cross member, frequently by the interposition of springs, so as to cushion its front end, in this manner permitting the rear axle to oscillate a trifle clockwise or counter-clockwise, under the influence of the torque reaction, in order to lessen the "jerks" due to sudden application of the load.

A number of torque arms were shown with the description of rear axles (see Figs. 274 and 351); others will be given later in this chapter.

In figuring the strength of torque arms it is advisable to consider the maximum stress which is likely to occur, and this takes place when the brakes lock the wheels, or when the clutch is suddenly thrown in, and the wheels slide on the ground. In the Hotchkiss drive, the spring resists the torque reaction; when the car is propelled forward the front half of the rear spring takes the torque reaction, and it is therefore flexed more than the rear half of the spring. When the brakes are applied, the tendency for the housing is to turn in the same direction as the wheels, and therefore the rear half of the spring is resisting the torque.

The coefficient of friction between rubber tires and dry road surface is usually taken at .6 (see page 85 for coefficient of friction); that is to say, every 1,000-lb. load on the rear wheel will create a pull on its periphery of $1{,}000 \times .6 = 600$ lb. If R is the radius of, and W the total load on, the rear wheels, the maximum torque T exerted on the rear axle housing when the wheels slide is, $T = W \times .6 \times R$ lb.-inches.

This torque is resisted by the torque arm, and if L is the length (from the center of the axle to the front end of the arm) and w the force or load at the front end, evidently $W \times .6 \times R = Lw$, which is the maximum bending moment in the torque arm. (See Bending moment of cantilever beams, page 17.) If the total load on both rear wheels were 2,500 lb. and the radius of the wheels 18 in., then the torque T created by the wheels when they slide on the ground (when the wheels are locked by the brake or when the clutch is suddenly thrown in and the wheels slip) equals $2{,}500 \times .6 \times 18 = 27{,}000$ lb.-inches, and this must be resisted by the torque arm.

If the torque arm length L is 50 in. (measured from the center of the axle) the maximum load w at its front end $= \frac{27{,}000}{50} =$ 540 lb. When a spring is used at the front end it must be made strong enough to support this maximum load. The maximum bending moment in the torque arm occurs where it is attached to the housing, and if this is 8 in. from the center of the rear axle, the length of the arm is $50 - 8 = 42$ in. Hence the maximum bending moment at this section of the torque arm is $42 \times 540 =$ 22,680 lb.-inches. (At various distances from the front end, the bending moment B, will, of course, vary. Suppose it is desired to find B, 15 in. from the front end, then $B = 15w = 15 \times 540 = 8{,}100$ lb.-inches.)

Knowing the moment of inertia I, or the section modulus Z, of the torque arm at any point, the stress in the metal may easily be determined. For instance, if a steel tube, of, say 2 in. in diameter, with a wall thickness of .25 in. were attached to the axle housing, the stress in the metal may be found from the formula $S = \frac{B}{Z}$, for a tube $Z = \frac{\pi}{4} \frac{(R^4 - r^4)}{R}$; R, the outer radius (of the tube just mentioned) is 1 in. and r, the inner radius .75 in.; hence $Z = \frac{\pi}{4} \frac{(1^4 - .75^4)}{1} = .536$.

If the maximum bending moment B is 22,680 lb.-inches, as found, the stress $S = \frac{22,680}{.536} = 42,300$ lb. per square inch approximately. This would be too high a stress in the metal unless its elastic limit were of about 140,000 lb. per square inch. Since we considered the maximum bending moment which is likely to occur, a factor of safety of 3½ (on the elastic limit) may be employed. If the ultimate strength is considered, a factor of safety of 4 may be used. If the steel tubing has an ultimate strength of, let us say, 90,000 lb. per square inch, a working stress of $\frac{90}{4} = 22,500$ lb. per square inch may be employed. Hence the tube before mentioned is too weak, and we may either use a tube of heavier section or larger diameter, or what is better, strengthen it at the rear, from the point on where the stress exceeds 22,500 lb. per square inch. From the equation $S = \frac{B}{Z}$ we have $B = SZ$, and substituting for S and Z, we have $B = 22,500 \times .536 = 12,000$ lb.-inches approximately; hence the permissible bending moment for the before mentioned tube must not exceed 12,000 lb.-inches. We must therefore find the portion of its length which must be strengthened. We saw before, $w = 540$ lb., and since $B = Lw$, $L = \frac{B}{w} = \frac{12,000}{.540} = 22$ in. approximately. In other words, B, 22 in. from the front end of the torque tube will be 12,000 lb.-inches and this corresponds to a stress S in the metal at that point of 22,500 lb. per square inch, for $\frac{B}{Z} = \frac{12,000}{.536} = 22,500$ lb.

If the torque arm is made of channel section as in the Packard car, (Fig. 271), the stress can be determined when we know the dimensions of the channel section at various points and then finding the section modulus. For example, suppose the section at a distance 15 in. from the front end of the torque arm to be 1 in. wide by 3 in. high, and the thickness of the metal .125 in.

We have assumed in this case a total torque arm length of 50 in., and a maximum torque of 27,000 lb.-inches, therefore the load at the front end is $\frac{27,000}{50} = 540$ lb.; 15 in. from the front, the bending moment will be $15 \times 540 = 8,100$ lb.-inches. The section modulus of a channel section $Z = \frac{bd^3 - h^3(b-t)}{6d}$, where b is the width; d the height; h the height less twice the thickness;

and t the thickness. (See page 77 for frame stresses.) In our case, therefore, $Z = \frac{1 \times 3^3 - 2.75^3\,(1 - .125)}{6 \times 3} = .488$. Therefore, $S = \frac{B}{Z} = \frac{8{,}100}{.488} = 16{,}700$ lb. per square inch. This gives a factor of safety of almost 5, if the ultimate strength of the metal is 80,000 lb. per square inch, and this is ample in this case.

In practice a variety of shapes and designs of torque arms are employed. In some designs the torque tube serves the purpose of propeller shaft housing as shown in Fig. 220, *i.e.*, a tube surrounds the propeller shaft and is rigidly attached to the rear axle housing; the front end extending to the universal joint behind the gear box. Sometimes the front end of the torque tube is forked, as shown in Fig. 391; the forks being rigidly attached to the rear end of the transmission case, while the universal joint is situated between the forks. This torque tube will also serve the purpose of radius and thrust rod, as the rear axle is kept at a given distance by this torque tube. With such a design both the front and rear ends of the rear spring should be shackled.

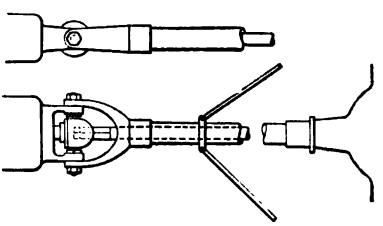


FIG. 391.—Torque tube.

To stiffen the construction, rods t (called tie-rods or hound-rods) may be used, which are attached to the outside of the torque tube, ordinarily near the front end, while the rear end is attached to the rear axle housing at widely separated points. The purpose of these rods is to strengthen the housing where joined to the torque tube, for when one wheel hits an obstruction and tends to move backward, it produces a great strain on the housing. Naturally, the axle housing may be made strong enough to withstand these strains without tie-rods, as is usually done in practice.

In Fig. 220 the propeller shaft, which forms the forward end of the bevel pinion shaft, is enclosed in a torque tube. The pinion is backed by a single row bearing with a single outer race; this bearing is subjected to purely radial loads. The end thrust, due to the bevel gear, is transferred by the pinion drive shaft (here also the propeller shaft) to the double row bearing mounted

at the front end of the torque tube; in this case a new departure double row ball bearing with both the inner and outer race clamped.

Figure 392 shows a construction where the front end of the torque arm is resting on coil springs, so that when driving forward, the torque reaction arising from the drive, will cause the torque arm to compress the upper spring, while when the brakes are applied, the lower spring will be compressed, and in this manner the action will not be so severe or harsh when starting or stopping the motor vehicle. The upper end *u* of the front torque arm support is pivoted to a bracket, attached to a cross member, at the side of the propeller shaft. As the front end in this construction is not rigidly attached to the frame, it will not prevent backward or forward motion of the torque arm, therefore separate radius rods must be used, unless the springs are made to take the thrust, *i.e.*, the front ends of the rear springs must be attached to the frame without shackles.

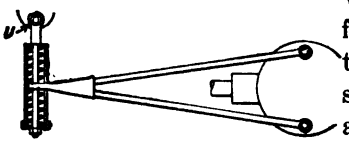


FIG. 392.—Torque arm.

Sometimes the torque arms are made of channel section, tapering toward the front as in the Packard car, (see Fig. 271). In this case, as the bending moment increases, the section modulus increases, and as the strain or the stress in the rod only acts in a vertical direction, a torque arm of channel section can be made lighter than one made tubular.

Instead of the front end being provided with a hole for a bolt as shown in Fig. 392, it may have a ball-and-socket connection like that shown in Fig. 271.

Thrust Rods.—The thrust rod usually also performs the duty of radius rod. It pushes the motor vehicle forward, hence the rod acts as a strut or a long column. Whenever the length of a column or a rod is more than 10 times its diameter, it is called a strut. In such event we cannot consider the compressive strength of the metal, since the rod will not be, or remain, perfectly straight, and when it is slightly bent it will buckle long before the elastic limit (in compression) of the material is reached.

The formula which the author finds useful in figuring the size of thrust rods is: $W = \left(M - N\frac{L}{d}\right)A$[1]; where W is the buckling

[1] See GOODMAN, *Mechanics*.

load; M a constant depending on the material; N a constant depending upon the form of the strut section; L the length of the strut in inches; d the smallest dimension, (diameter, height or width) across the section of the strut, in inches; A the area of the strut in square inches.

TABLE OF CONSTANTS

Material	Shape of strut	M	N	$\frac{L}{d}$ not to exceed
Mild steel.....	Square section	71,000	1,570	30
Mild steel.....	Round section	71,000	1,700	30
Mild steel.....	Tubular section	73,000	1,430	30
Mild steel.....	Channel or I-beam section	71,000	1,870	30
Hard steel.....	Square section	114,000	3,200	30
Hard steel.....	Round section	114,000	3,130	30
Hard steel.....	Tubular section	114,000	2,700	30
Hard steel.....	Channel or I-beam section	114,000	3,500	30

If, for instance, the thrust rod is 45 in. long and a round solid rod of mild steel is employed, it must not be less than 1.5 in. in diameter, since $\frac{L}{d} = \frac{45}{d}$, must not exceed 30, as given in the table. The buckling load W of such a rod will be, $W = \left(M - N\frac{L}{d}\right)A = \left(71{,}000 - 1{,}700 \times \frac{45}{1.5}\right) \times 1.75 = 35{,}000$ lb. (The area of a rod 1½ in. in diameter is 1.75 sq. in.)

The maximum draw bar pull (tractive effort or thrust) required to push a car or truck forward may be calculated (as will be shown later) or it may be assumed to be 30 per cent of the total weight of the car or truck with its maximum load. If, for instance, the total weight of a truck is 16,000 lb., the maximum thrust or tractive effort (on low gear when going up a very steep hill) may be assumed to be $16{,}000 \times \frac{30}{100} = 4{,}800$ lb., and the thrust in each rod is one-half this amount, or 2,400 lb. If the thrust rod is inclined upward, or deviates from the horizontal line, which it usually does in practice, the exact stress in the rod may be found by dividing the thrust load by the cosine of the angle.

For example, if the angle is 25°, the total stress in each rod would be $\frac{2{,}400}{\cos 25°} = \frac{2{,}400}{.906} = 2{,}650$ lb. approximately. We may use a factor of safety of 10, hence the rod should have a minimum buckling strength of $2{,}650 \times 10 = 26{,}500$ lb. A high factor of safety is necessary, as the wheels may hit the curb when backing the truck. Since the ratio $\frac{L}{d}$ must not exceed 30, when a solid bar is used, the diameter $d = \frac{L}{30}$, and this is more than necessary. When a tube, I-beam or channel section is employed, whose area is considerably less than that of a solid bar of the same outside dimension, the section must be proportioned so that W works out to be not less than 26,500 lb.

Tractive Effort.—The tractive effort (draw-bar pull) may be found from the maximum engine torque T. If the total gear reduction between engine and rear wheels be designated by e, the total efficiency $\left(\frac{\text{per cent}}{100}\right)$ by f, and the radius of the rear wheels by r, then the draw-bar pull or the tractive effort at the periphery of the wheel $= \frac{T\,e\,f}{r}$. (The *tractive factor* is the tractive effort divided by the total weight of the motor vehicle.) For instance, if the maximum engine torque is 2,800 lb.-inches, the total reduction 50, the efficiency on the low gear 70 per cent, the wheel radius 20 in., the tractive effort $= \frac{2{,}800 \times 50 \times .70}{20} = 4{,}900$ lb. If the weight of the truck fully loaded is 16,000 lb., then the tractive effort for each pound of truck weight, that is to say, the tractive factor $= \frac{4{,}900}{16{,}000} = .306$ approximately.

In chain driven trucks there is an added pull on the rods due to the tension in the chains and this must be added to the thrust load W. Supposing a chain driven truck weighs 10,000 lb., and the maximum draw-bar pull or tractive effort is 3,000 lb., or 1,500 lb. at each rear wheel. If the large sprocket (on the wheel) has a pitch radius of 9 in., and the radius of the rear wheel is 18 in., then the pull or tension in the chain multiplied by the radius of the small sprocket is equal to the tractive effort of each rear wheel, multiplied by the wheel radius, hence the tension in the chain is $\frac{18 \times 1{,}500}{9} = 3{,}000$ lb. In other words, the torque

at the wheel center would be $18 \times 1{,}500 = 27{,}000$ lb.-inches, and torque divided by radius of sprocket $\frac{27{,}000}{9 \text{ in.}} = 3{,}000$ lb., gives the force at the sprocket pitch circle, *i.e.*, the tangential pull on the sprocket or the tension in the chain.

The torque in the jackshaft may also be determined from the maximum engine torque, and the maximum gear reduction between the engine and the small sprocket. For instance, if the maximum engine torque is 2,000 lb.-inches, the low gear reduction in the transmission 4, and in the jackshaft bevels 2½, or a total of $4 \times 2\frac{1}{2} = 10$, and assuming the mechanical efficiency to be 90 per cent or .9, then the torque in the jackshaft is $2{,}000 \times 10 \times .9 = 18{,}000$ lb.-inches, or 9,000 lb.-inches in each half of the jackshaft. If the pitch radius of the sprocket pinion be 3 in., then the tangential force at the sprocket pinion circumference is $\frac{9{,}000}{3} = 3{,}000$ lb., and this is the tension in the chain which should be added to the draw-bar pull of 1,500 lb. found before, when determining the stress in the rod.

Radius Rod.—The purpose of a radius rod is, as its name implies, to keep the rear axle always at a certain "distance"—to make it follow a circle with the radius rod as the radius—when the spring is flexed. It should be remembered in this connection, that if only one radius rod is used at the center, as for instance, when the torque tube performs that function, the rear axle will not remain parallel with the front axle when the rear spring on one side of the car is deflected more than the other. For this reason two radius rods are often used, one on each side; this, as a rule is the case, when the radius rods also take the thrust.

In locating front anchorages of torque rods and radius rods, it should be remembered that if these are in line, (across the frame) with the universal joint, it will eliminate an undue amount of sliding motion in the propeller shaft joint when the axle rises and falls, by virtue of spring flexure.

Figure 393 is a detail of the radius rod and brake shoe assembly of the Mack 5,- 6½- and 7½- ton chain driven truck. The rod proper is of channel section, stamped, of ⅜-in. stock, gradually increasing in height from the front to the rear, as shown. With chain driven trucks, it is necessary to provide some means for shortening or lengthening this rod for the purpose of tightening the chain, or when required, to increase or decrease the chain

length. The brake shoes are anchored to the radius rod at *A*, hence the rod will take the reaction of the brakes. The front end of this rod is a ball, which, resting in a socket, forms a ball-and-socket joint, thereby being free to move in all directions.

In some designs, like that shown in Fig. 394, which is the radius (and thrust) rod of the Kelly-Springfield worm driven truck, the rear end of the rod is arranged to swivel vertically as well as

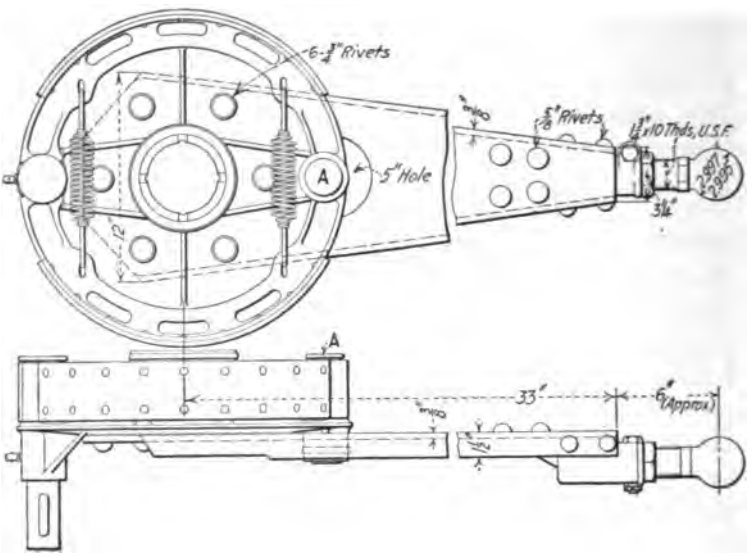


FIG. 393.—Mack (chain driven) truck radius rod and brake shoe assembly.

horizontally, (a universal joint effect), while the front end has provisions for vertical oscillation only. In this design the torque is taken through the springs and the thrust through these radius rods of I-section.

Truss Rods.—In some rear axles truss rods are employed, as shown in Figs. 323 and 325 to relieve the center of the housing of the deflection induced by the load on the spring pads. To calculate the truss rod tension, we must first know the load in the center of the axle. If we consider the housing a continuous beam supported at the ends and in the center, and the spring pads *s*, placed at a distance from the wheel of one-third the distance *L*, between wheel and the center of the axle housing (which approximates practical conditions) (see Fig. 395), we find, from the theory of beams, that the reaction *H* in the center is very

nearly one-half the total weight resting on both spring pads. We neglect the weight of the rear axle, and we assume that the center of the axle will suffer no deflection.

If the weight on each spring pad is ½ W; h the height of the

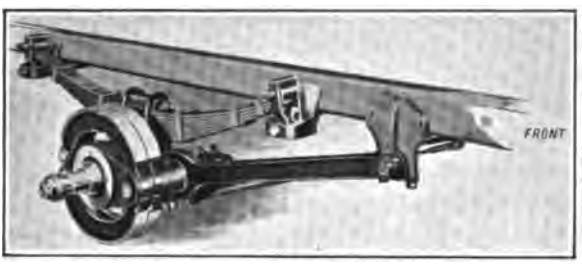


FIG. 394.—Kelly-Springfield (worm driven) truck radius and thrust rod.

vertical distance or the "dip" in the truss rod measured from the height of its end anchorage; t the length of each half of the rod and H the reaction in the center of the axle housing, the tension T

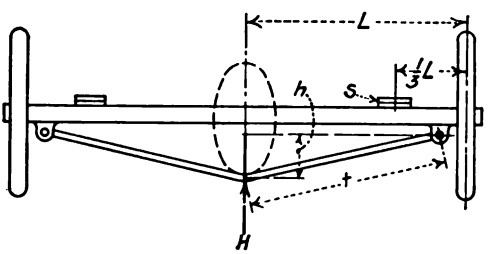


FIG. 395.—Truss rod.

in each half of the truss rod can be found by direct proportion.— $h:t::½H:T$, from which $h\ T = t\ ½H$, and $T = \frac{t \times ½H}{h}$ If h is 5 in., t 20 in. and $½H$, 500 lb., by substituting figures we have; $T = \frac{20 \times 500}{5} = 2{,}000$ lb. Allowing a factor of safety of 6, the strength of the weakest part of the rod must be figured on the basis of $6 \times 2{,}000 = 12{,}000$ lb. (Note, ½ of H is used in the calculations as only ½ of the reaction is supported by each half of the tie-rod.)

CHAPTER XI

BRAKES

In all self-propelled vehicles means must be provided for quickly and efficiently stopping the vehicles, and this is accomplished by applying friction to brake drums fitted to the rear wheels, and sometimes also to the propeller shaft, or in chain driven motor vehicles to the jackshaft.

Almost all motor vehicles are provided with two sets of independent brakes, one operated by the right foot pedal and called the service brake, and the other by a hand lever, called the emergency brake. In most American passenger cars, one set of brakes is applied to the exterior of the wheel brake drums, which are termed "external contracting brakes," another set, the "internal expanding brakes," are fitted to the interior of said brake drums.

Frequently, in trucks, both sets of brakes are of the internal expanding type, in which case the drum is made wider to accommodate two brake shoes or brake bands, side-by-side, on its inner side. A number of manufacturers, more especially in Europe, make use of the propeller shaft brake (in lieu of one set of wheel brakes) in which a brake drum is fitted to the rear of the transmission case. The advantage of braking on drums on the wheels is that the stress due to the retarding action of the wheels is confined to the wheel structure, whereas with the propeller shaft brake, the maximum stress is transmitted through the shafts and the gears of the final drive, the propeller shaft, etc., in other words, through all the parts between the wheel and the propeller shaft brake.

When it is considered that the braking effort is ordinarily much greater than the driving torque, this holds good especially in trucks of large capacity, where the torque, due to the locking of the wheels may be much higher than the maximum motor torque, it means that when the propeller shaft brake is used, all the parts between it and the wheel have to be designed to take care of this increased stress. The propeller shaft brake is the

most efficient for a given amount of pedal pressure, since the propeller shaft revolves at a higher speed than the rear wheels, therefore a much smaller effort is sufficient to lock the wheels.

Equalizers are often used on brakes, while more often they are dispensed with. It is claimed that if the brakes are properly designed and adjusted, the equalizers are unnecessary. It is, of course, a simpler and less expensive construction, but many manufacturers advocate equalizers as more efficient, since they equalize the braking effort on both wheels.

If a body is in motion, it possesses a certain amount of energy, and the same amount of energy must be expended in stopping such body. The energy stored in a moving body is called the kinetic energy. Hence if a motor vehicle is moving at a certain speed or velocity, with the power disconnected, it will possess a certain amount of kinetic energy which must be dissipated or used up, before it can come to rest.

If W is the weight of the moving vehicle in pound; V the velocity in feet per second; g the acceleration due to gravity, (equals 32.2 ft. per second, per second) its kinetic energy $K = \frac{W \times V^2}{2g}$. If sufficient time be given to the said moving vehicle, it will come to rest by itself, as the energy is absorbed by the friction in the bearings, the rolling resistance of the tires on the ground and the air resistance.

However, when the vehicle is stopped in a shorter time, an additional amount of energy must be expended to bring it to rest. This additional energy is usually expended by friction through brakes on brake drums, while the force used to apply such brakes is supplied by the driver through pedals or levers.

The brakes used on motor vehicles must be sufficiently powerful to lock the wheels, in which case the force tending to stop the machine will be that due to the force of friction between the rear wheels and the ground (neglecting for the present the rolling resistance of the front tires, friction of the moving parts and air resistance) and this equals the coefficient of friction multiplied by the load carried by the wheels which are locked.

For example, if an automobile weighing 3,000 lb. is moving at 30 ft. per second (about 20 miles per hour) its kinetic energy $K = \frac{W \times V^2}{2g} = \frac{3{,}000 \times 30^2}{2 \times 3.2,\dot{V}} = \frac{2{,}700{,}000}{64.4} = 42{,}000$ lb.-feet. If the speed of the motor vehicle is given in N miles per hour (and

since 1 mile = 5,280 ft., and one hour = 60 × 60 = 3,600 seconds) $V^2 = \left(N\frac{5,280}{3,600}\right)^2 = (1.467N)^2 = 1.467^2N^2 = 2.152N^2$. Hence $K = \frac{W\ 2.152N^2}{2 \times 32.2}$. Since $\frac{2.152}{64.4} = .0334$, therefore, $K = .0334WN^2$ lb.-feet. of energy stored in the car.

If the coefficient of friction c between tires and ground, when the wheels are locked, is 0.6, and w the weight resting on the rear wheels (or the weight on the wheels to which the brakes are fitted) then the force F in pounds at the periphery of the tires tending to retard or stop the car is $cw = 0.6 \times w$, and this force multiplied by the minimum distance or space S in which the car is stopped, equals the kinetic energy K, which is stored in the moving car by virtue of its velocity.

Hence, we also have $K = .6wS = .0334WN^2$, from which $S = \frac{.0334WN^2}{.6w}$. The fraction $\frac{.0334}{.6} = \frac{1}{18}$ approximately, therefore $S = \frac{WN^2}{18w}$. This formula can be simplified still further if $\frac{W}{w}$ is the ratio n of the total weight of the motor vehicle to the weight on the rear wheels, the formula becomes $S = \frac{nN^2}{18}$. If the total weight, for example is 3,600 lb., and ⅔, or 2,400 lb. of this rested on the rear wheels, then $n = \frac{3,600}{2,400} = 1.5$. If the vehicle were travelling at 20 miles per hour, the distance in which it could be stopped $S = \frac{1.5N^2}{18} = \frac{N^2}{12} = \frac{400}{12} = 33.3$ ft.

In this formula it is assumed that the coefficient of friction is 0.6. This, however, is not always the case, although the author believes it to be a good average; it may be as high as 0.75 with tires having rubber projections (anti-skid) or even higher when chains are attached, while when the tires and the road surface are smooth, it may be lower than 0.5 and considerably less if the road is wet or muddy.

In the above formula, we also neglect the rolling resistance of front tires on ground, and the losses due to the friction of the front wheel bearings and air resistance; their effect is to still further decrease the distance in which the car can be stopped when the wheels are locked. It was stated before that the force of friction

F at the periphery of the tires tending to stop the wheel from rotating $= cw$, and we know that the (moment of a force or) torque $T = F \times d$ (see Chap. XX, Vol. I) where d is the length from the point of application of the force to the axis around which it tends to turn, in this case it is the radius of the wheel. Hence $T = Fd = cwd$; for instance, if the radius of the wheel is 18 in., the coefficient of friction 0.6, and the load w resting on each rear wheel 1,200 lb., then $T = Fd = 0.6 \times 1{,}200 \times 18 = 12{,}960$ lb.-inches, (*i.e.*, the torque created in the wheel center by a force of $0.6 \times 1{,}200$ lb. pulling at a distance of 18 in. from the center of the wheel).

In order to lock the wheel this torque must be resisted, and this is accomplished by applying friction or a resisting force F_1 on the brake drum of radius d_1. When the wheels are locked $F_1 d_1 = Fd$ (*i.e.*, the force on the periphery of the wheel multiplied by its radius is equal to the force on the periphery of the brake drum multiplied by its radius) and this equals the torque T. From this formula, $F_1 = \frac{Fd}{d_1} = \frac{12{,}960}{d_1}$. If the radius d_1 of the brake drum is 9 in., the force on the brake drum to lock the wheels must be at least $\frac{12{,}960}{9} = 1{,}440$ lb. This is the retarding force and it is created by friction between the brake drum and the friction material. If the coefficient of friction between this material and the steel drum is 0.25, the actual (normal) pressure necessary between brake lining and the drum is $\frac{1{,}440}{.25} = 5{,}760$ lb.

This is distributed over the entire surface of the drum or over a portion of it. The greater the surface of the drum or the friction material coming in contact with the drum, the less will be the pressure per square inch of material, and, everything else being equal, its life will be longer.

The wheels are, as a rule, not entirely stopped by the brakes, hence there will be rubbing friction between drum and friction material, which will cause wear. The greater the radius of the brake drum, the smaller is the necessary force F_1 to lock the wheel. If in the example before given the friction material on the brake drum is 2 in. wide, the surface area on each drum equals the circumference multiplied by the width $= \pi d_1 \times 2$ in. $= 3.1416 \times (2 \times 9$ in.$) \times 2 = 113$ sq. in., and $\frac{5{,}760}{113} =$

51 lb. per square inch. (Brake drum diameter = 2 × 9 in.). In this example we assumed that the brake lining covered the entire drum surface.

The practice by manufacturers of announcing in their literature that their cars are equipped with a certain brake area, is no indication of the efficiency of the braking system. Unless it is known how far this brake area is located from the center of the wheel, it is impossible to find the effort required or the necessary pressure between brake drum and brake lining to lock the wheels. If the brake drum area in two cases is the same, but the drum diameter in one case is larger than in the other, the larger brake drum will require a smaller force F_1 and a smaller pressure between drum and lining to stop the wheel, and as the radiating surface of the drum is larger it will remain cooler, hence the lining will wear longer and if the operating linkage is the same, the pedal pressure is less.

The author thinks, therefore, that a much better indication of brake efficiency would be to state the required pressure per square inch between brake lining and brake drum to lock the wheels, when the car is traveling on the level, fully loaded.

When brake shoes are used it is advisable that they be provided with a liner of softer metal than the brake drum. If the brake drum is of steel, liners of cast iron or bronze are often used. Raybestos or asbestos friction liners are most commonly used in this country for the rear wheel brakes. Rivet heads in brake linings should set at least $\frac{1}{32}$ in. below the surface of the lining. Brake shoes are often provided with ribs for cooling purposes as well as for additional strength.

A number of manufacturers proportion the brake area, *i.e.*, the actual area of contact between the brake lining and the drum, as follows:—

Seventeen pounds of car weight per square inch of brake lining area for light and medium weight cars; 20 lb. for heavier cars; 30 to 50 lb. for medium and heavy trucks.

Brake Rod Linkage Layout.—In laying out brake rods, it should be kept in mind that the levers and rods must be so arranged that when the springs are flexed and the axle moves upward, the distances will remain the same, else the brakes might come into action automatically, when the vehicle travels over bad roads.[1] As to linkage leverage, for the hand brake on the rear

[1] See W. C. BAKER, *S.A.E. Transactions*, 1919, part 1, Correct Location of Brake Levers.

wheel brakes, it has been found satisfactory to use a leverage of 50 to 1; for the foot brake on the rear wheels, 20 to 1; for the transmission or propeller shaft brake it is made 26 to 1.[1]

[1] See paper by J. EDWARD SCHIPPER in *S.A.E. Journal*, April, 1922.

CHAPTER XII

FRONT AXLES

The front axle supports the front of the frame, as a rule by two semi-elliptic springs, which are attached to the axle at their centers and at their ends to the frame side members. Since the load upon the axle is transmitted through these springs, the greatest bending moment in the axle will occur at the spring seats, and it will be constant between the two springs. From the spring seats to the axle ends, the bending moment gradually decreases, as will be shown later.

The front axle, must, in addition, accommodate the steering knuckles, which are pivotally connected with the axle ends by means of which the motor vehicle may be steered. The front wheels are mounted on shafts or spindles forming a part of the steering knuckles, hence turning the knuckles to the right or the left will swing the front wheels in the same direction.

Steering Layout.—It should be remembered that when turning a corner it is necessary that both the front wheels as well as the rear wheels roll on the circumferences of circles having a common center. This requires that the two front wheels be interconnected in such a manner that the inner wheel (that nearest the center of the circle) describe a smaller circle than the outer wheel. Ideal steering conditions are obtained when no sliding action takes place between any of the wheels and the road surface when steering in either direction and describing any desired circle, large or small.

The matter is difficult of solution because the wheels do not always travel on a perfectly flat road, and when one wheel mounts an obstruction, or when one spring is deflected and not the other, conditions vary.

Another factor influencing steering conditions, is the rear axle, which is not always parallel to the front axle, for if one of the rear springs is deflected and not the other, in many constructions, the rear axle will not remain parallel to the front axle. To take all these conditions into account and design a layout theoretically

correct would require a complicated steering mechanism with linkages and a multiplicity of levers. However, by means of a single tie-rod interconnecting an arm of each steering knuckle, the theoretical conditions can be closely approximated, even though

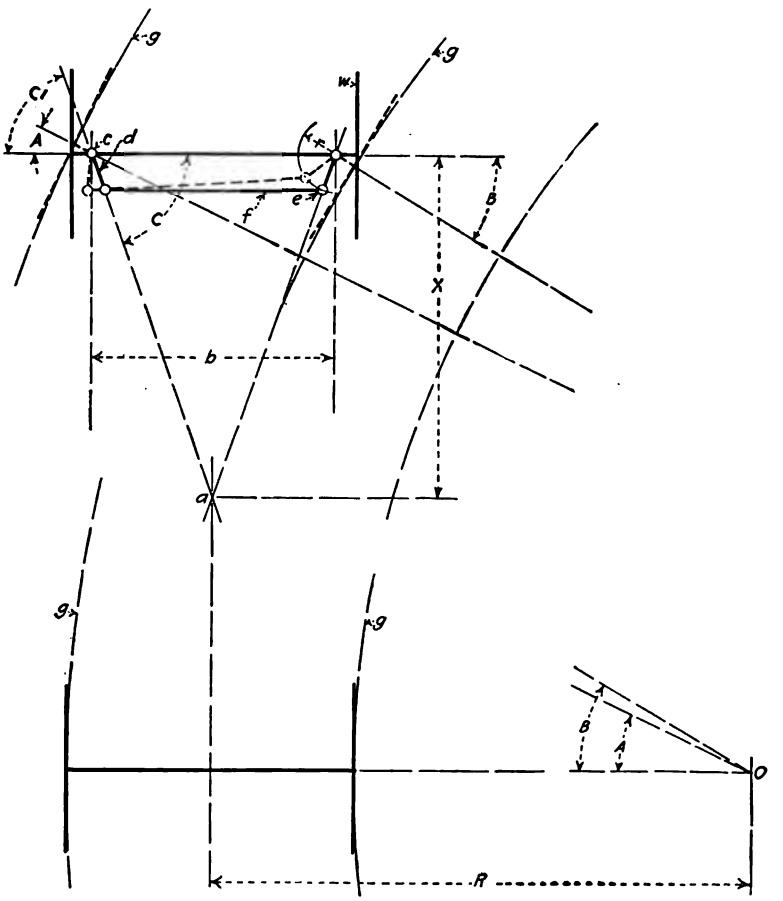


FIG. 396.—Steering gear layout.

they may not be absolutely correct, so that the front wheels turn very nearly around a common center under all conditions.

To approximate the theoretical conditions in practice requires that the arm of the steering knuckle project horizontally at a certain angle with respect to the wheel axle, so as to compensate for the difference in the circles described by the two front wheels whether steering to the right or the left, and whether the circles

be small or large. While a number of formulae have been worked out, some more or less involved, to determine the most favorable angle of the knuckle arms for given lengths of arms and given turning radii, the method described and shown in Figs. 396 and

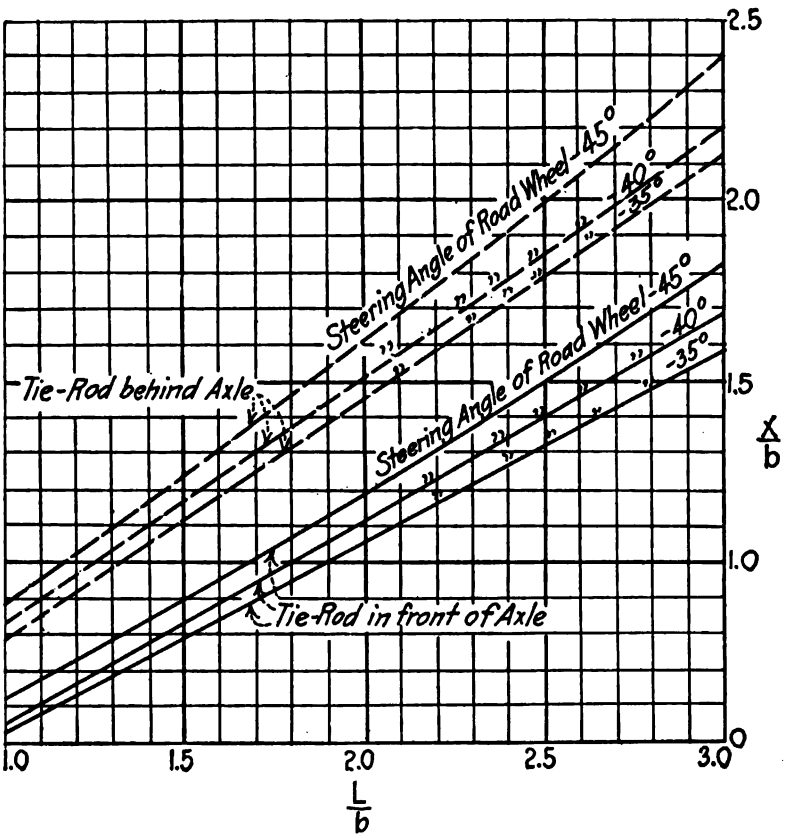


FIG. 397.—Steering gear curves.

397 is very accurate and can readily be employed in practice. It is found that for only one turning radius are the conditions strictly accurate, while for others there will be a slight error. When designing steering gears, it should be remembered that the maximum angle of the front wheel when making a turn should not, as a rule, exceed about 38° or 40°, for if it does, the car, as it moves forward, will tend to slide the front wheels sidewise and the rear wheels will not follow the front wheels. The more rapid the car speed, and the lower the coefficient of friction

between front wheels and road surface (when it is slippery the coefficient is very low), the greater is the tendency for the front wheels to skid sidewise when steering at an angle.

The dotted lines of the front wheels, tie rod and knuckle arms, shows their positions when steering around a circle having a common center C, and it may be noted that the inner wheel, due to the angle of the steering knuckle arms, will be deflected to a greater angle than the outer wheel, in order to describe a smaller circle. The knuckle arm shown dotted on the left, is practically at right angles to the tie-rod (when the wheels are in the position as shown), while the knuckle arm (dotted) on the right forms a greater angle; in this position a comparatively small movement of the left knuckle arm will impart a larger motion to the right steering knuckle (and to the wheel) and vice versa; when turning to the left, the right arm will be nearer a right angle with respect to the tie-rod and the left wheel will be deflected to a greater degree. In this figure are also shown circles g, representing the paths described by the four wheels, when steering around center O.

A and B are the angles formed between a line drawn through the axis of the rear wheels and steering pivot centers of the outer and the inner front wheels respectively. b is the distance between the steering pivots c. C is the angle formed between the steering arms and the front axle, whether the arm be located in the front or in the back of the axle, as seen from the figure. d is the steering arm of length r, f the tie-rod or cross-rod. X is the distance between the front axle and the point of intersection a, which determines the angle C of the steering arms with respect to the front axle.

As stated previously, the steering conditions will be absolutely correct when the circles g described by the four wheels turn about the common center O. Correct steering conditions are fulfilled when

$$\cot A = \frac{R + \frac{b}{2}}{L}$$

and

$$\cot B = \frac{R - \frac{b}{2}}{L};$$

hence

$$\cot A - \cot B = \frac{b}{L}.$$

This matter was thoroughly investigated by Lutz, in the

"Motor Wagen" of October, 1908, and he found the results as given in Fig. 397 and the table given later.

The requirement is that the steering mechanism determine the correct angle A of the outer wheel for a corresponding angle B of the inner wheel. The smaller the total cramping angle (steering angle) of the wheels, the less is the error involved between the correct angles A and B. Calculations were made for three wheel steering angles, 35°, 40° and 45°.

In the curves, the ratio $\frac{r}{b}$ was taken as .14, but were checked for values of from .11 to .17, without showing any appreciable errors. The results are plotted for values of $\frac{L}{b}$, *i.e.*, wheel base over distance between steering pivot centers, between 1 and 3, with the corresponding values of $\frac{X}{b}$. For instance, let us assume the maximum steering angle of A or B to be 40°. If the wheel base L is 112 in. and the distance b between steering knuckles 50 in., $\frac{L}{b} = \frac{112}{50} = 2.4$. Assuming the tie-rod to be behind the front axle, as is most frequently the case, by looking at the curve we see that for a ratio $\frac{L}{b}$ of 2.4 we find the value of $\frac{X}{b}$ to be 1.78 approximately. The distance b is fixed and let us say it is 50 in. then, since $\frac{X}{b} = 1.78$, $X = 1.78 \times b = 89$ in.

Having found the location of point a in Fig. 396 to be 89 in. ($= X$) distant from the front axle, it is easy to determine the angle C of the steering arms by drawing a straight line from steering pivot centers c to a. In this manner the steering mechanism can be laid out correctly with the aid of these curves without the use of a large drawing board or without drawing the layout on a small scale in order to have center O on the drawing board.

The length r of the steering arms (from the center of the pivot to the center of the eye) must not be made too small, for the shorter it is, the greater is the strain in the tie-rod and the joints, and if any play develops the error is greater. This length varies from about 4 in. in small touring cars to about 9 in. in large trucks, the length being measured from the center of the steering-knuckle c to the center of the steering arm eye e. When the knuckle arms and the tie-rod are in front of the axle,

they are usually made somewhat longer and they are bent outward, toward the wheel, but its angle C' is the same as C, when the tie-rod is behind the front axle.

The following table gives the various angles for different proportions of $\frac{L}{b}$, when the steering angles of the wheels are 35°, 40° and 45°:

If $\frac{L}{b}$ =	Tie-rod in front and steering angle of			Tie-rod behind and steering angle of		
	35°	40°	45°	35°	40°	45°
	C' =	C' =	C' =	C =	C =	C =
1.5	58°	59°	60°	66°	67°	68°
2.0	65°	66°	67°	71.5°	72°	73°
2.5	69.5°	71°	72°	74.5°	75°	76.5°
3.0	72.5°	74°	75°	77°	78°	79°

Camber of Front Wheels.—A further requirement for ideal steering is that the front wheels should so function that when they are swung to the right or to the left, when the car is standing still, the contact points between wheel and ground should remain the same; this will facilitate ease of steering as the wheels will not have to be moved forward or backward when they swing around. To obtain such a condition necessitates in practice that the bottom of the front wheels touch the ground at a point intersecting with a line through the axis of the knuckle pin, and this requires the front wheel spindle to be tilted downward at an angle A (see Fig. 398), to give the wheel a certain camber. However, tilting it to such a degree as shown here would impart a great strain to the wheel and impair the appearance; for this reason in practice the angle A is made only about 2° or 3°. On the other hand, the more accurately the wheel contacts the ground at the point where it meets a line through the axis of the knuckle pin, the less will be the strain on the pin under normal running.

Caster Effect of Front Wheels.—Sometimes the knuckle pin, when viewed from the side of the car, is tilted back slightly on the top, for which is claimed a caster effect, *i.e.*, the wheels will naturally tend to assume a "straight forward" position, and this

renders steering easier. In this construction the knuckle arm ends and the tie-rod move vertically, (up and down) as well as horizontally when steering.

Toeing In.—(See Fig. 437*b*.) In most automobiles and trucks the front wheels are "toed in" in the front, from ¼ to ½ in., *i.e.*, the distance between the wheels in the front, on a horizontal line with the axis, is made from ¼ to ½ in. less than at the back. The reason for giving the wheels this "toe in" is that the normal tendency of the wheels is to spread out in the front when the car is in motion. By means of the tie-rod end connections the front wheels are given this "toe in," when their natural tendency to

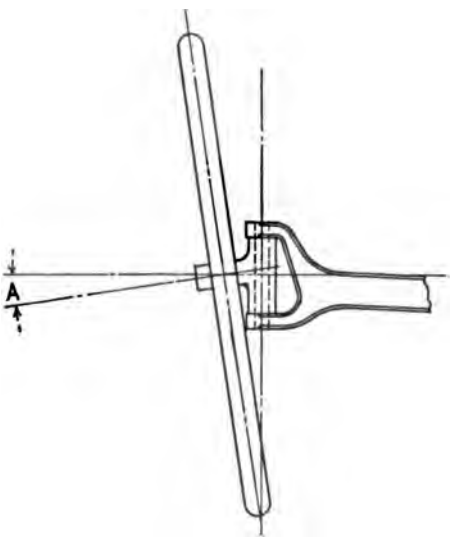


FIG. 398.—Camber of front wheel.

spread is compensated by the friction between the wheel and the ground. The combined "toeing in" and "cambering" reduces the effort required for steering.

Front Axle Loads and Axle Section.—The load on the front axle is ordinarily vertical, although when hitting obstructions the horizontal stress may be quite severe, depending on the height of the obstruction and the speed. Front axles are usually made of I-section, which is best suited to withstand vertical but not horizontal loads. Round front axles, which were formerly more in evidence than now, can resist stresses equally well in all directions; however, they must be made heavier for a given vertical load. The present I-section axles, together with their

ends, are usually drop forged in a single piece. In most instances the spring pads are also forged integral, and the axle is bent in the center to allow more clearance under the crankcase. The designer should consider the possibility of front spring breakage and allow sufficient clearance that the axle may not hit the crankcase in such event, with consequent damage to the latter. In front axles it is advisable to allow a factor of safety of at least 6, when considering the maximum vertical static load, (full load when the car is at rest); with alloy steels a higher factor of safety is usually employed.

Occasionally the front axle is so attached to the front springs (with intervening spring pads) that the entire axle is slightly tilted back on the top. In such position, the maximum strength of the axle is not entirely in a vertical line, but in a line slightly inclined forward at the bottom and backward at the top, and this is the best condition in practice when the wheel hits an obstruction. To obtain sufficient horizontal rigidity, practice has demonstrated that the height of the section should not be more than 1.5 times the width; judging from a number of successful models which the author examined, the average ratio of height over width seems to be 1.35. The thickness of the center web should not be less than about one-sixth of the width.

Front axles are usually forged with a draft angle of from 7° to 10°, (see Fig. 399), or slightly less. In drop forgings a draft angle is necessary to prevent the object from sticking in the dies, as the forging contracts when it cools. However, it is possible to reduce the draft angle on the outside of the top and bottom flanges slightly and still have the forging clear the die sufficiently when the forging cools. This increases the axle strength, as a larger area of metal is at the top and at the bottom, *i.e.*, farther away from the neutral axis.

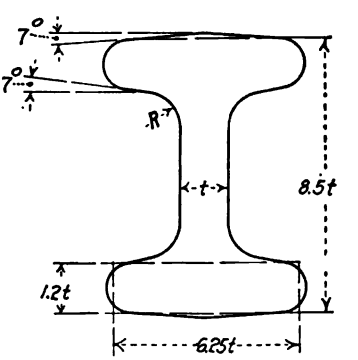


FIG. 399.—Front axle section.

Figure 399 shows approximately the average proportion of front axle section as used in a number of motor vehicles. The thickness of the center web being denoted by t, the other dimensions are given in terms of t. The radius R varies between $\frac{3}{16}$ in. and $\frac{3}{8}$ in.

Stresses in Front Axles.—As an example, take Fig. 439, which is the axle of the eight-cylinder Oldsmobile. From the section *A-A* it is seen that the overall height is 2¼ in., width 1⅝ in., thickness of center web 7/32 in., radius of the edge of top and bottom flanges ⅛ in. Assume that when this car is fully loaded, a weight of 1,600 lb. is resting on the two front wheels or 800 lb. on each. The tread is 56 in. and the distance between spring centers 28½ in. Thus $56 - 28.5 = 27.5$, and $\frac{27.5}{2} = 13.75$ in which is the distance between the spring center and the front wheel center.

Considering the front axle as a simple beam, with two concentrated loads resting upon it, as shown in Fig. 22, the reaction at each wheel is, as was stated, 800 lb., hence the bending moment underneath the spring is $13.75 \times 800 = 11{,}000$ lb. inches approximately, and it is constant between the two springs.

To determine the stress in the axle metal we have the formula

$$S \text{ (stress per square inch)} = B \frac{(\textit{bending moment in pound-inches})}{Z \textit{ (section modulus)}}$$

For an I-beam or channel section Z equals the section modulus of the solid, rectangular section (as found from the overall dimensions) minus the section modulus of the cutaway section (the portion missing on the inside).

If d is the height, b the width of the section, h the height between the flanges and w the width minus the center web (which $= b - r$, see Fig. 82) $Z = \frac{b\,d^3 - h^3\,w}{6 \times d}$. By looking at section *A-A*, Fig. 439, determine the average dimensions (not the overall dimensions) as indicated in Fig. 399. In order to obtain a fairly accurate value for the section modulus, we find $d = 2\frac{3}{16}$ in., $b = 1\frac{1}{2}$ in., $h = 1\frac{11}{16}$ in., $w = 1\frac{9}{32}$ in., hence

$$Z = \frac{1\frac{1}{2} \times 2\frac{3}{16}^3 - 1\frac{11}{16}^3 \times 1\frac{9}{32}}{6 \times 2\frac{3}{16}} =$$

$$\frac{1.5 \times 2.1875^3 - 1.687^3 \times 1.281}{6 \times 2.1875} = \frac{1.5 \times 10.45 - 4.8 \times 1.281}{13.12} =$$

$$\frac{15.7 - 6.15}{13.12} = \frac{9.55}{13.12} = .73 \text{ approximately,}$$

and

$$S = \frac{B}{Z} = \frac{11{,}000}{.73} = 15{,}000 \text{ lb. per square inch approximately,}$$

Axles are frequently made of 1035 S.A.E. steel and are heat treated to a scleroscope hardness of between 35 and 40, when the metal has a tensile strength of about 95,000 lb. per square inch and an elastic limit of about 65,000 lb. per square inch. Hence, under static load (when at rest) the stress in the front axle metal is 15,000 lb. per square inch and the factor of safety with regard to its ultimate strength is $\frac{95,000}{15,000} = 6.33$.

The author believes it is better to consider the elastic limit of any material, as it is a truer indication of the lasting qualities of a structure. The factor of safety with respect to the elastic limit is $\frac{65,000}{15,000} = 4.33$, which is none too much, when it is considered that the wheels are subjected to severe shocks when striking obstructions on the road.

Steering Head.—Front axles, in many designs, are provided with forked ends, the steering knuckle being placed between the yoke as illustrated in Figs. 401 and 441. This construction is usually called the Elliott type. In the reversed Elliot type of steering head, the steering knuckle is forked and accommodates the end of the axle between its yokes (see Figs. 438 and 440). As a rule a pin or shaft, called the steering knuckle pin or king pin, passes through the knuckle and the axle head, thereby establishing a pivotal connection between these two, and allowing the steering knuckle to swivel in the axle head. Occasionally the steering knuckle is provided with integral pivots which extend into the bearings of the axle head, thus dispensing with the knuckle pin, as in Fig. 431. When a knuckle pin is employed, as is the usual practice, it may swivel in the axle head and be rigidly attached to the knuckle, (as is ordinarily the case in the Elliott type), or it may be rigidly attached to the axle yokes and rock in bearings in the knuckle. The inverted Elliott type usually requires a greater distance between the wheel and the knuckle pin, which would suggest a greater bending moment and hence a greater stress in the knuckle for a given metal section; on the other hand, as the knuckle is yoked, the bearing surfaces are farther apart, and this tends to reduce the stress in the knuckle forks.

Another construction sometimes used is the Lemoine type, where the steering knuckle is inserted into a plain (single, not forked) axle head, either from the top or from the bottom, one example of which is described in Fig. 436.

In addition to the knuckle arms which are connected to the tie rod, another arm is attached to one of the steering knuckles for the purpose of establishing connection with an arm of the steering column, through the drag link (sometimes called the steering gear connecting rod). At times one arm of the knuckle is used for steering as well as for the tie rod while at other times separate arms are employed (see Fig. 437). Figure 400 shows the various types used by the Sheldon Axle & Spring Co. for their lines

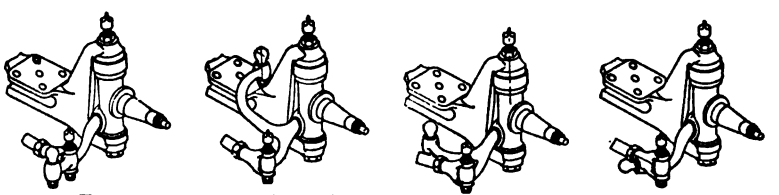


FIG. 400.—Steering knuckle arms (Sheldon Axle & Spring Co.).

of front axles. Some manufacturers forge the knuckle arms, integral with the knuckle, others attach these separately by means of tapered fittings, key and nut.

The greatest strain in the knuckle pin, and in the front wheel bearings, occurs when the front wheels skid or travel around a corner at a high speed. By referring to Figs. 389 and 390 and the description of the stresses in full-floating axles, pages 322 and 334, the student should have no difficulty in calculating the stresses in front wheels, steering knuckles, etc.

Figure 401 shows assembly drawings of the U.S.A., Class B truck front axle. A vertical section through the steering knuckle pin, etc., is seen on the left. Figure 402 is a detail of the axle forging, made of S.A.E. 1035 carbon steel, heat treated. The axle is of the regular Elliott type. The axle is of I-section, 4 in. high by 3 in. wide, with the center web from $\frac{1}{2}$ in. to $\frac{9}{16}$ in. thick. In the center the axle is curved to give additional crankcase clearance. Figure 403 is a detail of the left hand steering knuckle, made of S.A.E. 3130 steel, heat treated. Near the shoulder the knuckle is $2\frac{9}{16}$ in. in diameter and it is deflected downward 2° to give camber to the wheels for the purpose before mentioned. The heat treatment and the hardness of the steel are noted on the drawing. Figure 404 shows the assembly of the bushings in the steering knuckle. The stud passing through *s* holds the steering knuckle pin in position.

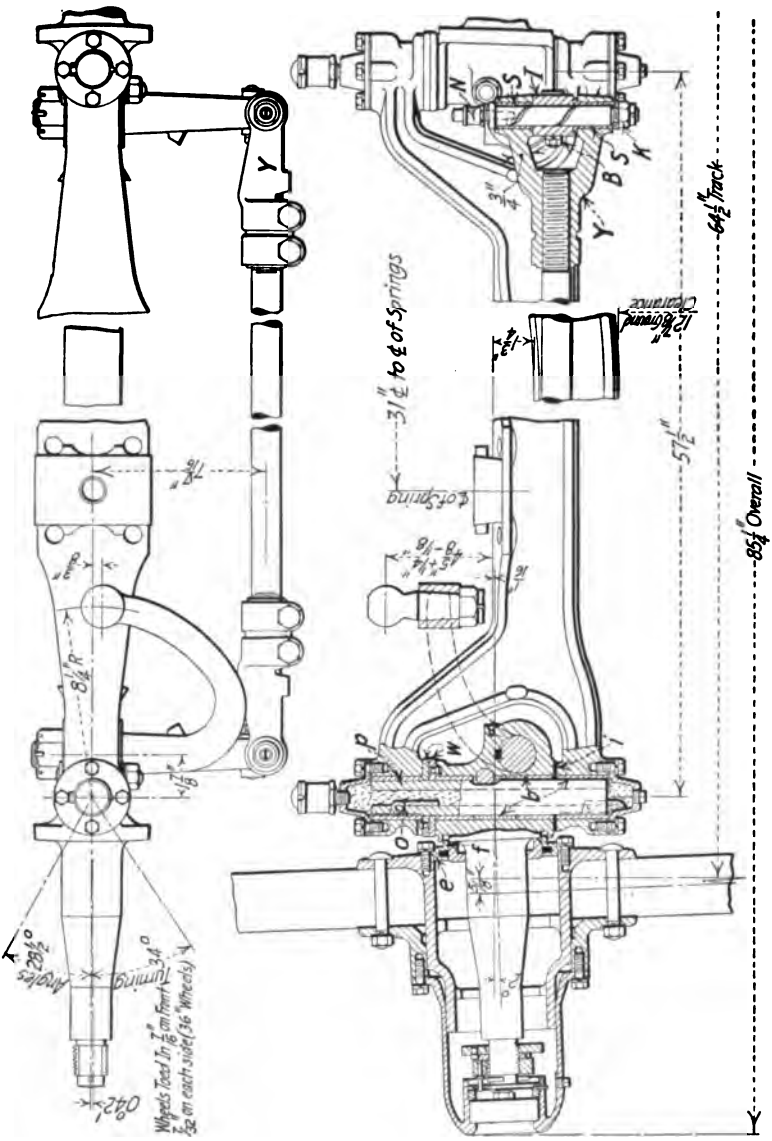


FIG. 401.—Class B truck front axle.

Even though this pin is stationary in the knuckle, steel bushings *b* are provided for the purpose of eliminating wear on the knuckle arising from blows on the wheel. This wear is greatest on the top and bottom where the knuckle bears against the knuckle pin, and in time the knuckle will wear near the ends as shown by dotted lines *w*. In the event of such wear, it is only required to change the bushings, whereas if there were no bushings the entire knuckle would have to be replaced.

Figures 405 and 406 are details of the front hub and flange. Both are made of malleable iron.

"The hub closure is designed with the object of presenting the smallest diameter for the felt rubbing face. A flange *f* (see assembly and Fig. 407) on the felt retainer, rotating within a groove in the knuckle, prevents the dirt from reaching the felt."[1] The felt is riding on the hub bearing spacer made of S.A.E. 1025 steel, either forged or made of screw stock. The felt retaining ring is bolted to the hub as shown by means of set screws and lock washers. The knuckle pin *p*, a detail of which is shown in Fig. 408, is made of carbon steel, also the bushings *b*; both are hardened and ground. The bushings are pressed into the ends of the axle forging. The pin is hollow and contains an oil wick *o* (see assembly) for the purpose of lubrication. It was found that ordinary grease commonly used for lubricating this pin is not entirely satisfactory; oil lubrication was adopted even though considerable leakage of oil was foreseen. The thrust washers *w* (see also Fig. 409) have oil grooves *g*, the oil entering them through recesses *r*, next to the pin. The steel washer *i* is made in various thicknesses, so that in the event of play or wear between knuckle and axle head, a thicker washer may be substituted.

The knuckle arms (Fig. 410 being the right hand arm) are forged to shape, except for a hand-bending operation on the ball arm. They are secured in the knuckles by tapered fits and keys. Figure 411 is the left hand arm. The tie-rod connecting the two arms together is attached to the end *T*, while steering is effected by arm *S*. Into this arm the steering arm ball, (Fig. 412), is fastened. The tie-rod is made of round stock 1¼ in. in diameter, and to its ends are attached the steering cross-rod yokes *Y* (see assembly). Bushings *S* (see detail in Fig. 413) made of S.A.E. 1020 steel, carbonized, are pressed into the forked ends of the

[1] See G. W. Carlson, *S.A.E. Transactions*, 1918, Part 1.

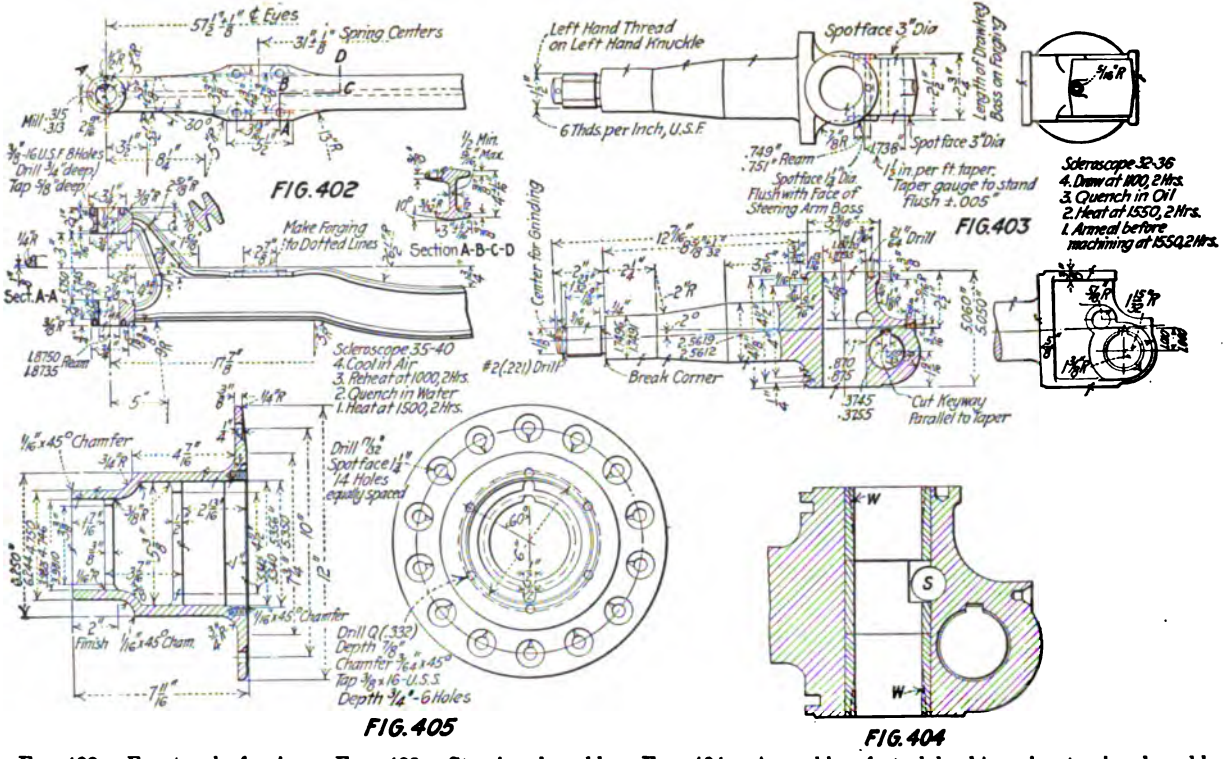


FIG. 402
FIG. 403
FIG. 404
FIG. 405
Sect. A-A
Section A-B-C-D
Make Forging to Dotted Lines
Sclerascope 35-40
4. Cool in Air
3. Reheat at 1000, 2 Hrs.
2. Quench in Water
1. Heat at 1500, 2 Hrs.
Left Hand Thread on Left Hand Knuckle
6 Thds. per Inch, U.S.F.
Spotface 3" Dia
Spotface 3" Dia
Length of Drawing Boss on Forging
Sclerascope 32-36
4. Draw at 1100, 2 Hrs.
3. Quench in Oil
2. Heat at 1550, 2 Hrs.
1. Anneal before machining at 1550, 2 Hrs.
Spotface 1½" Dia. Flush with Face of Steering Arm Boss
Taper gauge to stand Flush ±.005"
Break Corner
Cut Keyway Parallel to Taper
Drill 7/32" Spotface 1½" 14 Holes equally spaced
Drill Q (.332) Depth 7/8" Chamfer 7/64 x 45°
Tap 3/8 x 16-U.S.S. Depth 3/4" - 6 Holes
1/16" x 45° Chamfer
Finish 1/16" x 45° Cham.
W
S

Fig. 402.—Front axle forging. Fig. 403.—Steering knuckle. Fig. 404.—Assembly of steel bushings in steering knuckle. Fig. 405.—Front wheel hub (Class B truck).

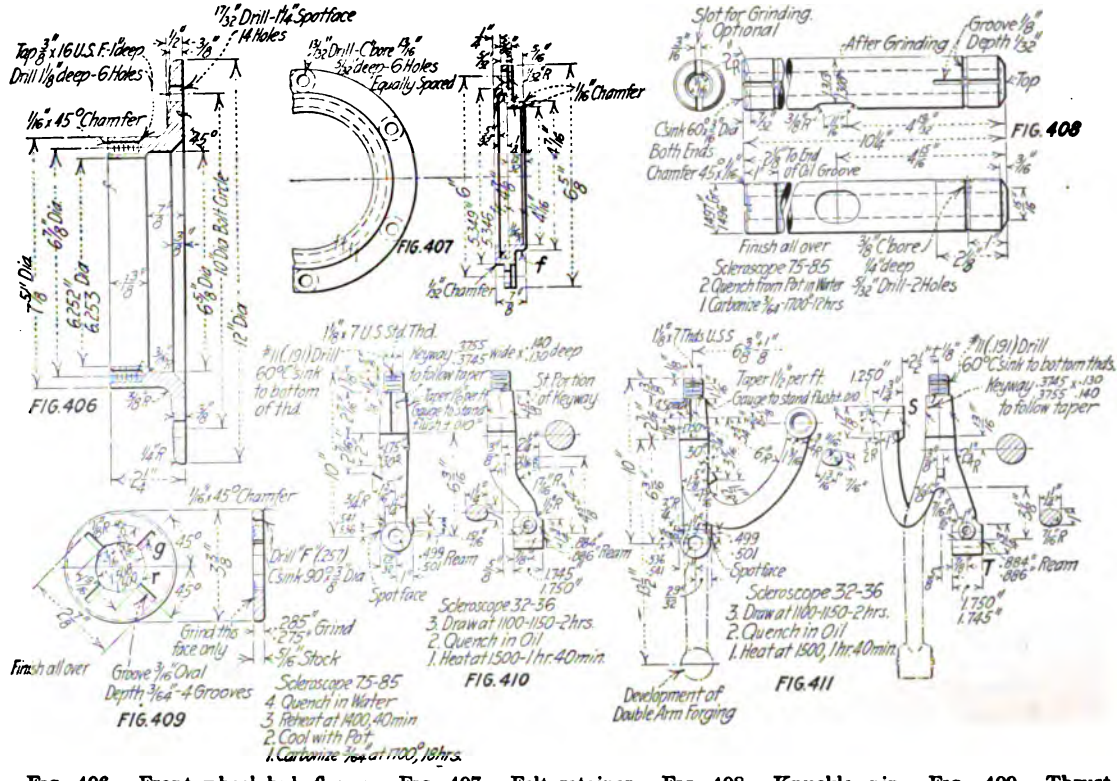


FIG. 406
FIG. 407
FIG. 408
FIG. 409
FIG. 410
FIG. 411
Development of Double Arm forging
Sclerascope 75-85
4. Quench in Water
3. Reheat at 1400, 40min
2. Cool with Pot
1. Carburize 3/64 at 1700°, 18hrs.
Sclerascope 32-36
3. Draw at 1100-1150-2hrs.
2. Quench in Oil
1. Heat at 1500-1 hr 40min.
Sclerascope 75-85
2. Quench from Pot in Water
1. Carburize 3/64 1700°-12hrs.
Finish all over
Grind this face only
Groove 3/16" Oval Depth 3/64" - 4 Grooves
Spot face
Taper 1" per ft. Gauge to stand flush 010
Keyway .3745 wide x .140 deep to follow taper
St. Portion of Keyway
Slot for Grinding. Optional
After Grinding
Groove 1/8" Depth 1/32"
Drill 1 1/8" deep - 6 Holes
Top 3/8" x 16 U.S.F. 1" deep
1/16" x 45° Chamfer
Ream
Finish all over

Fig. 406.—Front wheel hub flange. Fig. 407.—Felt retainer. Fig. 408.—Knuckle pin. Fig. 409.—Thrust washer. Fig. 410.—Knuckle arm. Fig. 411.—Steering arm (Class B truck).

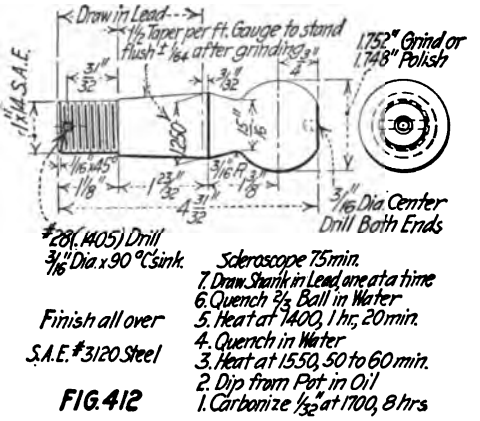

Draw in Lead
1/2 Taper per ft. Gauge to stand flush ± 1/64 after grinding
1.752" Grind or 1.748" Polish
1.744 S.A.E.
#28 (.1405) Drill
3/16" Dia x 90° C'sink.
3/16" Dia Center Drill Both Ends
Finish all over
S.A.E. #3120 Steel
Scleroscope 75 min.
7. Draw Shank in Lead one at a time
6. Quench 2/3 Ball in Water
5. Heat at 1400, 1 hr, 20 min.
4. Quench in Water
3. Heat at 1550, 50 to 60 min.
2. Dip from Pot in Oil
1. Carbonize 1/32" at 1700, 8 hrs.

FIG. 412

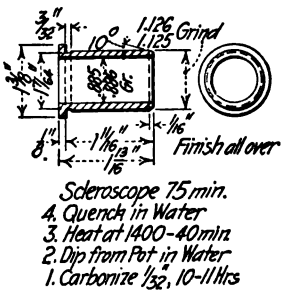


FIG. 413

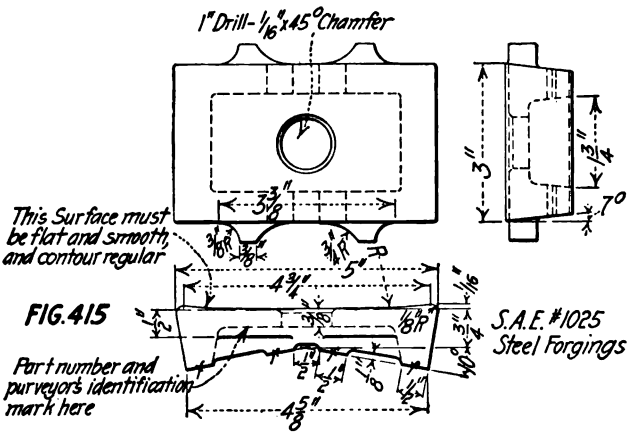

1" Drill- 1/16" x 45° Chamfer
This Surface must be flat and smooth, and contour regular
Part number and purveyor's identification mark here
S.A.E. #1025 Steel Forgings

FIG. 415

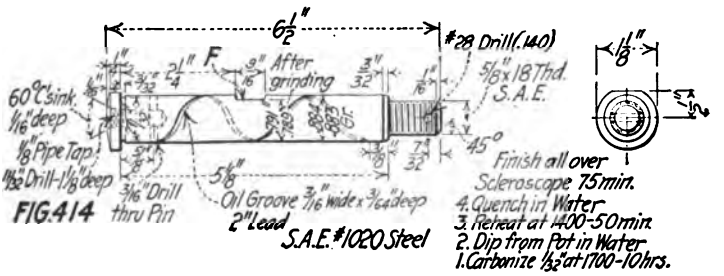


FIG. 414

FIG. 412.—Steering arm ball. FIG. 413.—Cross-rod yoke bushing. FIG. 414.—Cross-rod yoke pin. FIG. 415.—Spring seat.

yokes. Both bushings, the upper and the lower, have their flanges on the top, to provide a bearing surface between the flange of the lower bushing and the eye *I* of the steering knuckle arm, while the flange of the upper bushing bears against the head of the steering cross-rod end pin (Fig. 414), made of S.A.E. 1020 steel, carbonized. Note the spiral oil groove on the surface of this pin, into which oil is fed by means of an oiler *N*, (see assembly). These cross-rod end pins are prevented from turning by a "flat" *F*, and a stud *B*, hence there will only be friction between the pin and the hardened bushings in the cross-rod ends or yokes. The front axle is provided with spring seats, (Fig. 415), which are placed on the top of the axle spring pads. These spring seats are forgings made of S.A.E. 1025 steel.

Tie-rods or Cross-rods.—Sometimes the knuckle arms and the tie-rod are placed behind, and sometimes in front of the axle. When the rod is in front, it is more subject to damage in the event of an accident, and the ends of the rod must extend very closely to the front wheels, since the arms are bent outward when in front of the axle.

When steering to the right and left the tie-rod will be alternately in compression and tension, hence it is advisable not to bend this rod, for a bend cannot withstand compression so well. When the steering knuckle pin is vertical, the knuckle arms will move in a horizontal direction only, and therefore clevis or forked yoke ends may be used on the tie-rod, with a clevis bolt passing through them, as shown in Fig. 400. On the other hand, when the steering knuckle pin is at an angle, the knuckle arms will not only move horizontally, when steering to the right or left, but also vertically, hence ball and socket joints between tie-rod and knuckle arms are indispensable. Coil springs are placed behind the sockets, with ball and socket joints, to reduce vibrations arising from road shocks.

In some models, ball-and-socket joints are employed for tie-rod connections, even when the knuckle pins are straight, as they are claimed to give better satisfaction. With the clevis or forked type connection, the continuous vibrations are liable to act like small blows when there is any play, and this will wear out, or pound out, the bushings in the clevis or in the eye of the steering knuckle arms, while with the ball-and-socket joint, the spring behind the socket absorbs or softens such blows. In the Packard 3-ton truck (see CHURCH, *S.A.E. Transactions*,

1916, Part II) a ball 1⅝ in. in diameter and a spring pressure of 300 lb. has been found to give good satisfaction in the tie-rod end. Another advantage of the ball-and-socket is that there is no play between them, while such will exist sooner or later with a yoke and pin. Balls-and-sockets should be hardened and polished.

Timken Axle.—The Timken front axle for a truck of 3½-ton merchandise capacity is shown in Fig. 416. The chief difference

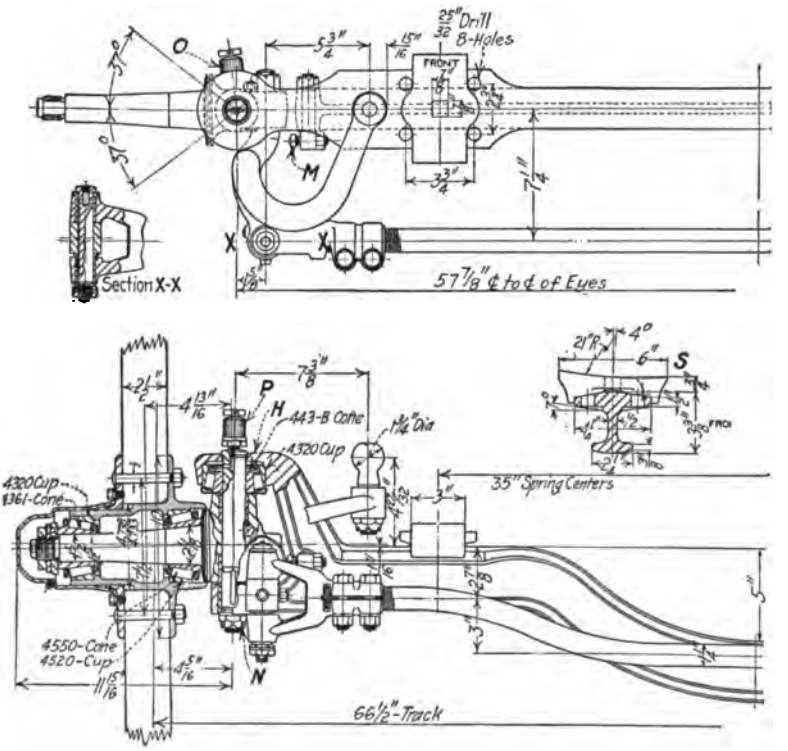


FIG. 416.—Timken front axle for 3½-ton truck.

between this model and the Class B axle, (which is also nominally of 3½-ton capacity, but is in reality as strong as a commercial 5-ton model) is in the axle head at *H*. Timken tapered roller bearings are provided on the knuckle head instead of plain friction thrust washers. A double clamp attaches the yoke to the cross-tube, the same as in the Class B axle. Through the shape of the spring seats, (see *S* in the sectional view), the bottom of the

front axle and knuckle pins are tilted forward, in order to produce a caster effect in steering. All the bearing surfaces, including axle spindles, steering knuckles, cross-tube yoke bolts and bushings, are ground accurately to size and held within limits of .001 in. By means of the nut *N* on the steering knuckle pins, the tapered roller thrust bearings can be adjusted to allow for wear. *M*, is an adjustable stop to regulate the amount of throw of the steering knuckle. All nuts are castellated and locked with cotter pins. Note the dust washers underneath the thrust bearing and how dust is excluded from the top by recessing the knuckle pin head into the upper axle head yoke; at the bottom the nut *N* extends into the lower axle yoke.

The cross-tube (tie-rod) yokes are provided with oil or grease cups and grooves, likewise the steering knuckles, as can be seen at *O* and *P*, for lubricating the knuckle pins.

The following table gives data of various Timken models:

DETAILS OF I-BEAM SECTION

"Merchandise" capacity, tons	Total height, inches	Total width, inches	Web thickness, inches	Wheel spindle diameter near shoulder, inches	Approximate weight complete without wheels, pounds
5	4	2½	⅝	2⅝	441
3½	3⅜	2¼	½	2⅛	264
1½	2⅞	2¼	7/16	1 13/32	175
¾–1	2¾	2⅛	5/16	1¾	150

In the two smaller models, thrust washers are used in the axle head instead of roller bearings, and the construction is similar to that of the Class B axle.

The cross-tube diameter in all the models is 1¼ in., except in the smallest size where it is 1 in.

The steering ball diameter in the 5-ton and 3½-ton models is 1¾ in. in in the two smaller models 1¼ in.

Sheldon Front Axle.—Sheldon front axles are equipped with either ball bearings, straight roller or tapered roller bearings, (Fig. 417). Front axles are one-piece forgings of carbon steel. Steering knuckles and steering levers are forged of chrome-nickel steel, annealed and heat treated. Tie-rod clevises are drop forged. Ball pins, knuckle pins (pivot bolts) and clevis pins are hardened and ground. The bearing seats on the wheel

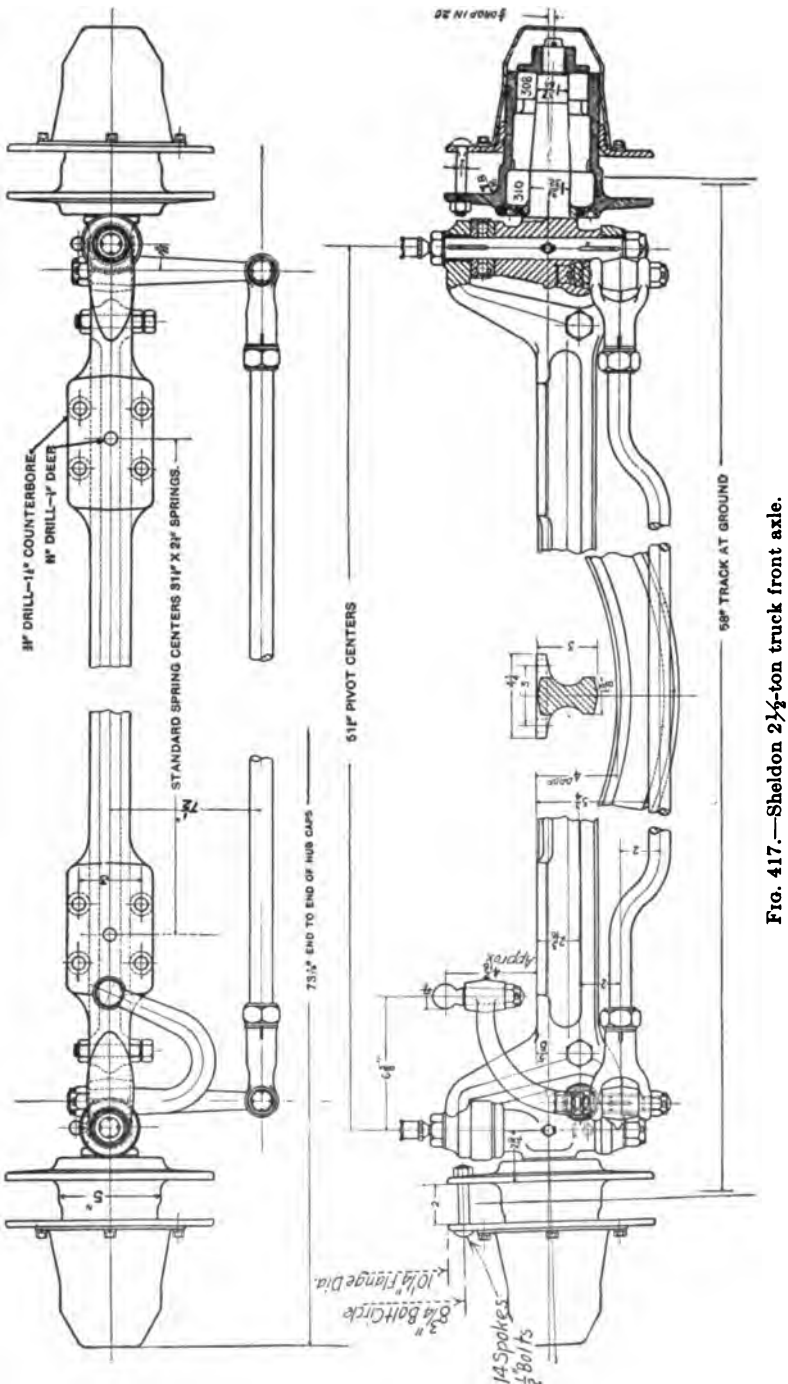


FIG. 417.—Sheldon 2½-ton truck front axle.

spindles are carefully gaged; these surfaces are held to limits of .0005 in. for outer bearing, .001 in. for inner bearings.

Bushings.—In the axle-head yokes and the knuckle arms, hardened steel bushings are provided. The wear between the knuckle and the yoke of the axle head is taken up, on the lighter models, by two hardened and ground thrust washers; on the heavier types, by ball thrust-bearings. The ball pin has a tapered shank which allows it to be drawn up tight in the event of wear. The pivot bolts (knuckle pins) are anchored to the steering knuckles and thus revolve in the steel bushed yoke of the axle end.

The Sheldon 2½-ton Truck Axle.—Figure 417 is an assembly drawing. Figure 418 is a detail of the front axle forging made of S.A.E 1035 steel. In all the 2½-, 3½- and 5-ton models, the Sheldon axles are not of ordinary I-beam section, but as shown in the figure. Figure 419 is a detail drawing of the steering knuckle made of 3135 chrome-nickel steel. Note that the threads on the axle spindle are right hand for the right hand knuckle and left hand for the left hand knuckle; this eliminates the tendency of the wheels, as they revolve forward, to loosen the nuts which hold the wheels on the axle spindles. Figure 420 is a detail of the pivot bolt, or steering knuckle pin, made of S.A.E. 1020 steel, carbonized (hardened) and ground to the limits indicated; only the threads are left soft. This pivot bolt is pinned fast to the steering knuckle by the hole at *P*. The upper bearing surface is lubricated by an oil cup screwed in the top of the head while the lower one by an oil cup screwed in the side of the knuckle at *O* (see Fig. 419). Figure 421 shows the two types of double steering levers, right and left (dotted, when attached to the left knuckle), the drag link being attached to one end *D*, and the cross-rod end pins to *E*. The dimension *A* for rear steering is made 0 in., while for front steering (tie-rod in front) this dimension is 2 in. *B* also varies in the various models between 0 in. and 2 in. *R* varies from 1 3/16 in. to 3⅞ in., while *Z* varies from 5⅜ in. to 6 in. The hole at *D* may be straight or .750-in. taper per foot, while the end *E* is furnished with a ball end if desired, as seen on the left, for connection with cross-rod ends having sockets.

Figure 422 is the ball pin used for the 2½-ton model; it is made of S.A.E. 3135 chrome-nickel steel, the ball being 1½ in. in diameter and polished to reduce friction. In the larger sizes the ball is 1¾ in. in diameter. The cross-rod is made of S.A.E.

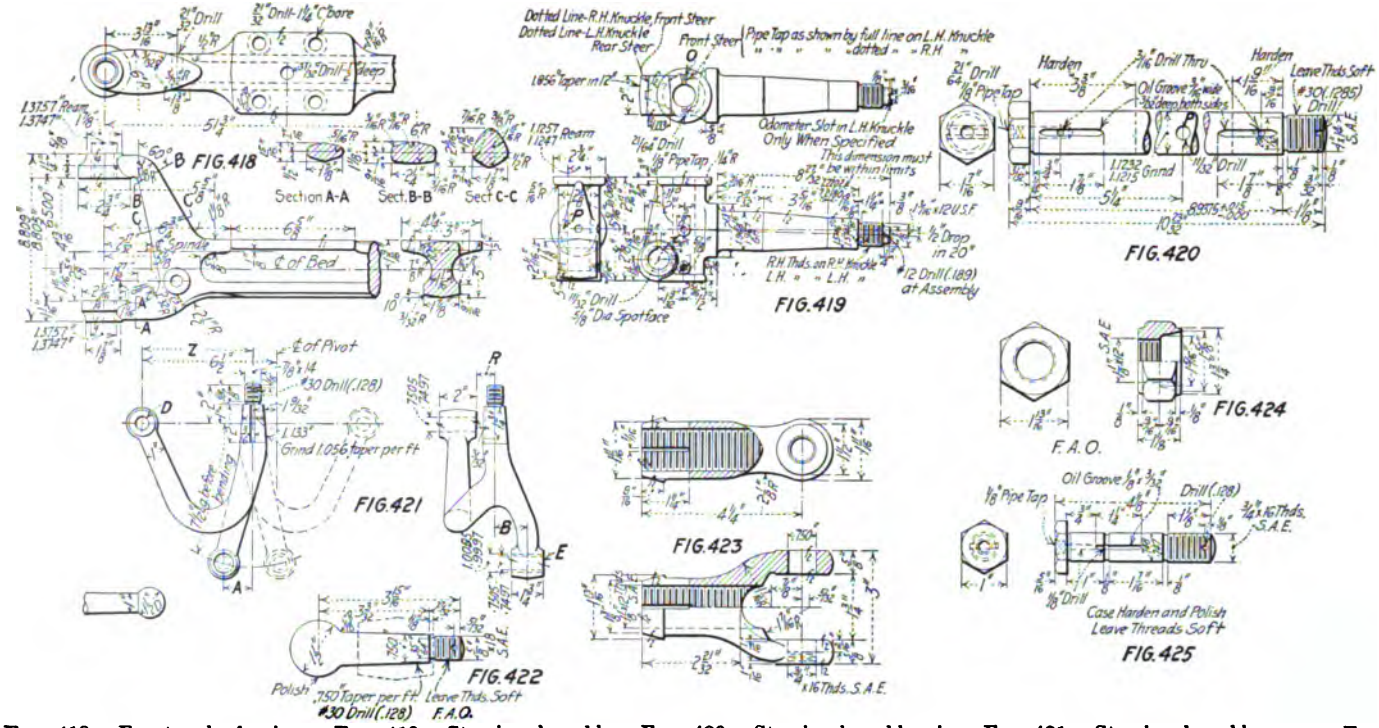


Fig. 418.—Front axle forging. Fig. 419.—Steering knuckle. Fig. 420.—Steering knuckle pin. Fig. 421.—Steering knuckle arms. Fig. 422.—Ball pin. Fig. 423.—Tie-rod clevis. Fig. 424.—Clevis jamb nut. Fig. 425.—Clevis pin (Sheldon axle, for 2½-ton truck).

1030 steel, 1⅛ in. in diameter, the ends being provided with standard S.A.E. 1⅛ in. by 12 threads. Figure 423 is a detail of the clevis or cross-rod end yoke, made of S.A.E. 1020 steel, drop forged. The clevis is threaded for attachment to the cross rod and slotted for a length of 1¼ in. It is locked tight on the rod by the clevis jamb nut, Fig. 424 (made of S.A.E. screw stock 1114), which is bored out at one end to fit over the taper of the clevis. Figure 425 is the clevis pin made of screw stock, case

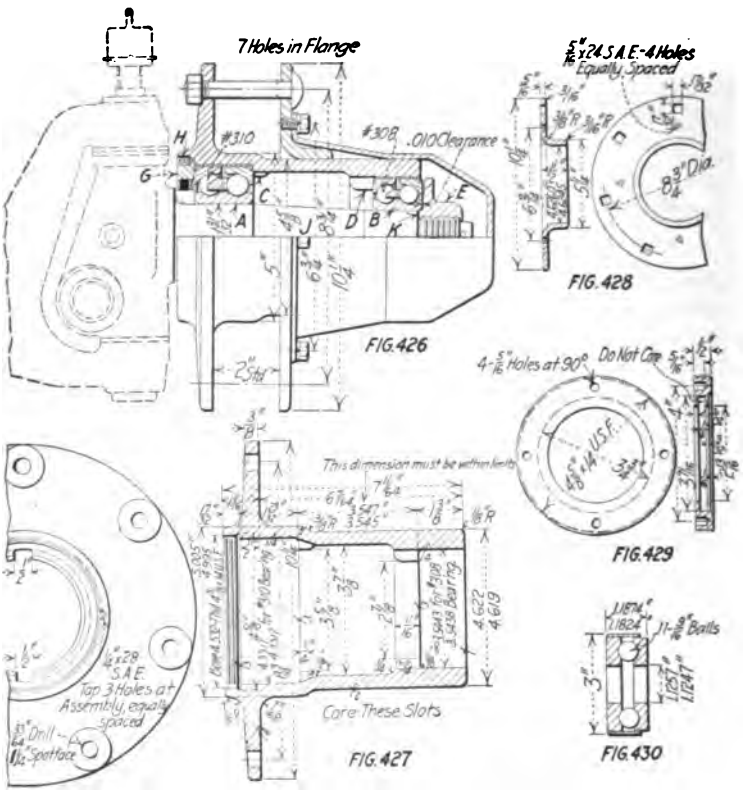


FIG. 426.—Front wheel hub assembly. FIG. 427.—Front wheel hub. FIG. 428.—Front hub flange. FIG. 429.—Bearing retainer. FIG. 430.—Thrust-bearing assembly (Sheldon 2½-ton truck axle).

hardened and polished, with the threads left soft. It is screwed into the lower clevis fork and oscillates in bushings in the cross-rod arm attached to the steering knuckle. The front hub assembly is shown in Fig. 426.

The following are the mounting directions for this assembly:

1. Insert bearing *A* and press against shoulder *C*.
2. Screw retainer *G* into place and lock with screw *H*.
3. Insert bearing *B* and press against shoulder *D*.
4. Mount hub on axle *J* and tighten nut *E* against shoulder *K*.
5. Very important that there is at least .010 in. clearance between nut *E* and bearing *B*.

Figure 427 is a detail of the front wheel hub, made of malleable iron; the number of bolt holes in the hub for attachment to the wheel vary with the number of spokes; bolts may be placed between each spoke or between every second spoke; in the latter case the number of holes will vary between 6 and 8. The pressed steel front hub flange is shown in Fig. 428. The holes are made square for carriage bolts while the smaller holes are provided for attachment of the front hub cap.

Figure 429 is the wheel bearing retainer (malleable iron) which is threaded on its periphery and attached to the wheel hub. It is provided with a felt ring groove, as seen from the assembly. Figure 430 shows the steering knuckle thrust bearing assembly with steel cup, to exclude dust from the bearing.

The following table gives some additional data of Sheldon axles:

Merchandise capacity of truck, tons	Details of front axle section			Wheel spindle diameter near shoulder, inches	Approximate weight complete (without wheels), pounds
	Height, inches	Width, inches	Web thickness		
5–6	4	2⅝	Almost solid	2 15/16	490
3½	3⅜	2¼	See Fig. 418	2⅜	340
2½	3	1⅞	See Fig. 418	1 31/32	220
1½	2¾	1¾	7/16 in.	1 31/32	186
1	2⅝	1¾	.....	1 9/16	135

Merchandise capacity, tons	Maximum load allowable on spring pads (front), pounds[1]	Spring centers, inches
5–6	7,000	36½
3½	4,880	33½
2½	3,800	31½
1½	3,000	29½
1	2,000	29

[1] On the two smaller sizes it is the maximum load allowed on the front tires.

Figure 431 is the front axle and steering rod assembly of the White ¾-ton truck, which is identical in construction with all the White models, except that metal wheels are furnished on the 3½-ton and 5-ton models. In the White front axle no knuckle pin is used but the knuckle itself is pivoted where it enters the upper and lower forks of the axle head. Bushings are provided

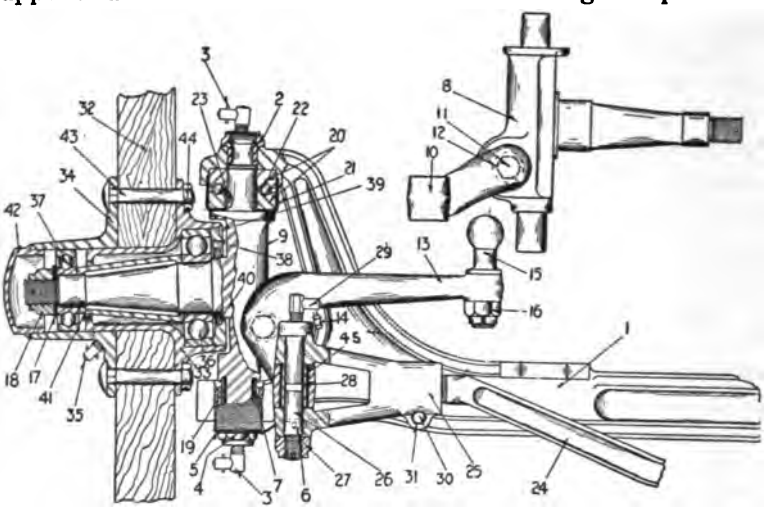


FIG. 431.—White truck front axle.

at all wearing surfaces as seen from the construction. The following table gives the names of the various parts of the complete front axle:

Ref. No.	Number Per Car	Name of Parts
1	1	Front axle.
2	2	Center upper bushing steering knuckle.
3	4	Oilers.
4	2	Front axle lower bushing.
	2	Front axle lower bushing binder bolts.
	2	Lock washers.
5	2	Lock washer.
6	2	Lock screw.
	2	Lock washer.
7	2	Pivot nut bushing.
8	1	R.H. steering knuckle.
9	1	L.H. steering knuckle.
10	1	Steering knuckle arm R.H.
11	2	Nut.
12	2	Washers.
	2	Key (No. 25 Woodruff).
13	1	Steering knuckle arm L.H.
14	1	Screw.

Ref. No.	Number Per Car	Name of Parts
15	1	Steering knuckle arm ball.
16	1	Nut.
17	2	Washer.
18	2	Nut.
19	2	Steering knuckle sleeve.
20	4	Cone.
21	2	Steering knuckle cone dust cap.
22	22	Steel ball (½-in. standard steel).
23	2	Key (No. 6 Woodruff).
24	1	Steering knuckle tie rod.
	1	Rod boot R.H.
	1	Rod boot L.H.
25	2	Tie rod ends.
26	2	Tie rod end pins.
27	2	Nut.
28	2	Rod end pin sleeves.
29	2	Grease oilers.
30	2	Bolt.
31	2	Nut.
32	2	Front wheel without hub.
33	2	Front wheel hub.
34	2	Front wheel hub flange.
35	2	Grease oiler.
36	2	Ball bearing.
37	2	Ball bearing.
38	2	Front wheel dust ring.
39	2	Retainer cap lock pins (⅛- × 1- in.).
40	2	Front wheel dust ring felt.
41	2	Front wheel bearing spacer.
42	2	Front wheel hub dust cap.
43	10	Bolts (¼- × 3- in.).
44	10	Nuts (¼- in.).
45	2	Steering knuckle bushing.

Figure 432 is an assembly view of the Pierce-Arrow 3½- and 5-ton truck axle. The steering knuckle is of the inverted Elliott type, with the knuckle pin larger in diameter in the top bearing than at the bottom. The steering arm *a* is attached to the knuckle by means of the stem of knuckle arm *b*. In the 2-ton model this company uses a steering knuckle pin of uniform diameter and the more conventional steering arm, consisting of the ordinary fork type, placed, with the tie rod, behind the front axle. In the 3½- and 5-ton models, on the other hand, the tie-rod is in the front and the steering arm connected to the drag link in the rear as shown.

By means of screw *c* shown dotted, the steering angle of the wheels is adjusted. A projection from the knuckle strikes the front end of this screw, thereby limiting the cramping angle.

In the ¾-ton Torbenson front axle, made of 1040 S.A.E. steel, heat treated, all draft angles of the I-beam are 7°. Details of the axle forging are seen in Fig. 433, which gives all the dimen-

sions. Figure 434 is a detail of the steering knuckle, made of 3135 S.A.E. steel. Figure 435 is a detail of the steering arm, made of the same material, annealed and heat treated similarly to the steering knuckle. The tie-rod used with this axle is 1 in. in diameter, tubular.

Figure 436 is the Marmon front axle, wheel hub, and steering knuckle. This knuckle is of the *L* head, Lemoine type, in which the knuckle is placed into the axle head from above and

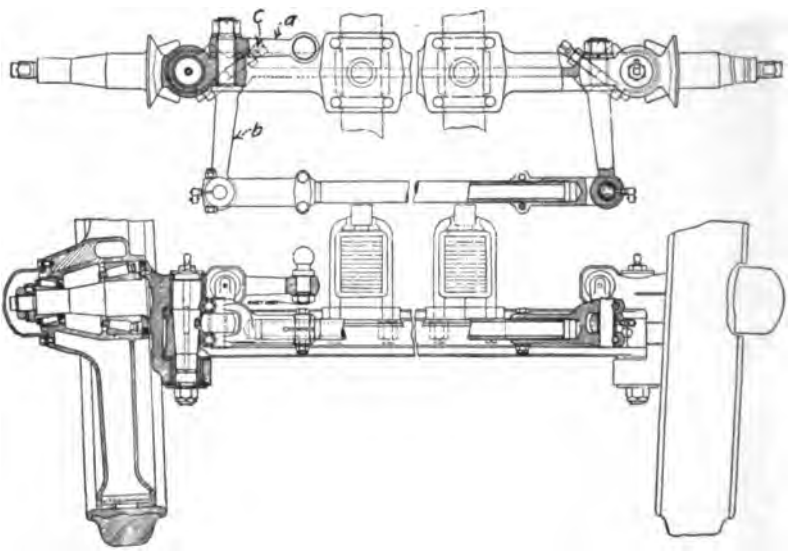


FIG. 432.—Pierce-Arrow 3½- and 5-ton truck axle.

held in position by a large nut at the bottom, which takes all the thrust through a ball bearing between it and the axle end; the axle is thus suspended from the knuckle spindle. This spindle revolves in bronze bushings in the axle head. The wheel hub is supported by tapered roller bearings. A felt packing on the inner side and a cap on the outside provide a grease tight housing for the bearings. The front wheel is "quick detachable" and is fitted on the outside of the hub as noted from the illustration.

In order to remove this hub it is required, first, to remove the wheel, which is accomplished by removing the hub cap; then the lock ring is removed and the felt washer retainer; then the bearing nut cap is taken off, the cotter pin, the nut and the tonged washer, after which the hub will slide off. By having the

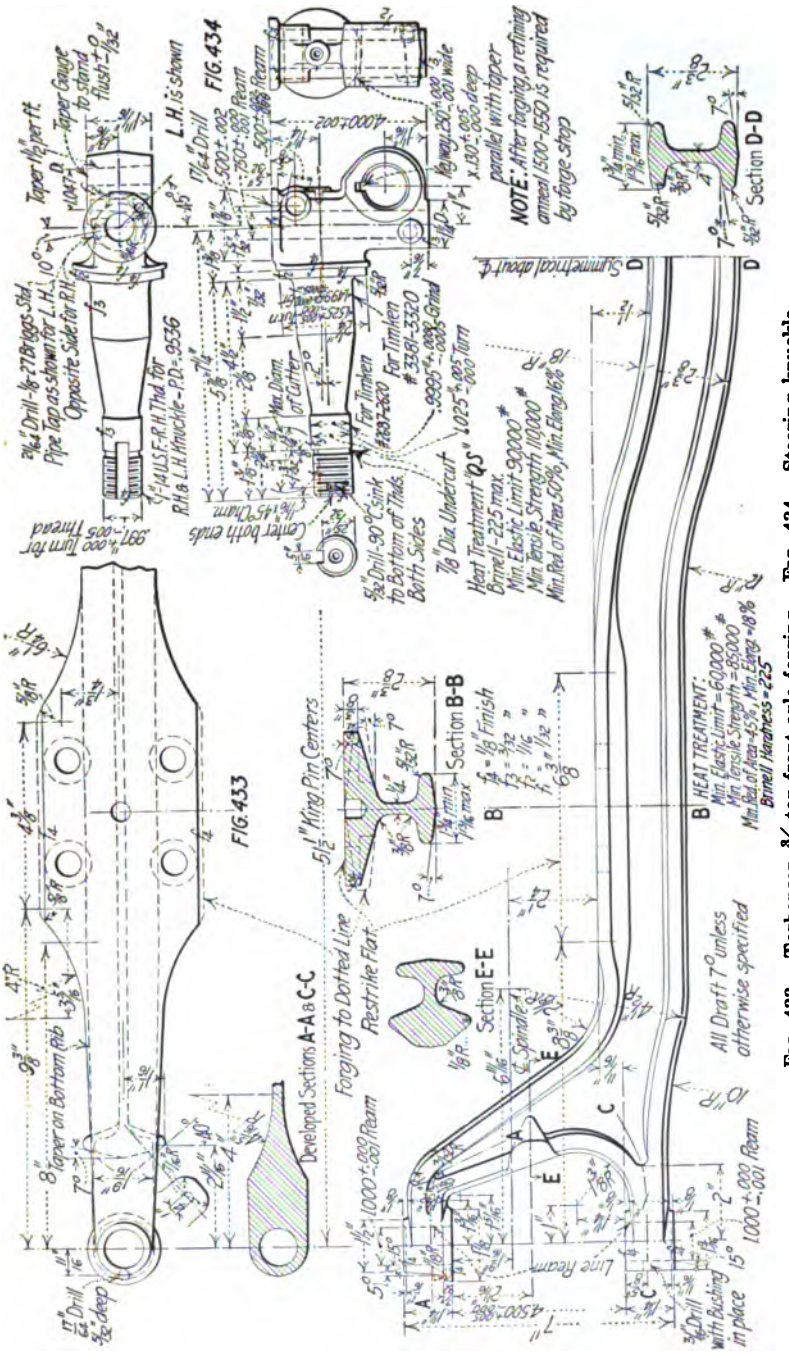


FIG. 433
FIG. 434
Developed Sections A-A & C-C
Section B-B
Section E-E
Section D-D
Forging to Dotted Line
Restrike Flat
All Draft 7° unless otherwise specified
HEAT TREATMENT:
Min. Elastic Limit = 60000 #
Min. Tensile Strength = 85000 #
Min. Red. of Area = 45%, Min. Elong. = 18%
Brinell Hardness = 225
L.H. is shown
NOTE: After forging, a refining anneal 1500-1550 is required by forge shop
Line Reams
Symmetrical about ℄

Fig. 433.—Torbensen ¾-ton front axle forging. Fig. 434.—Steering knuckle.

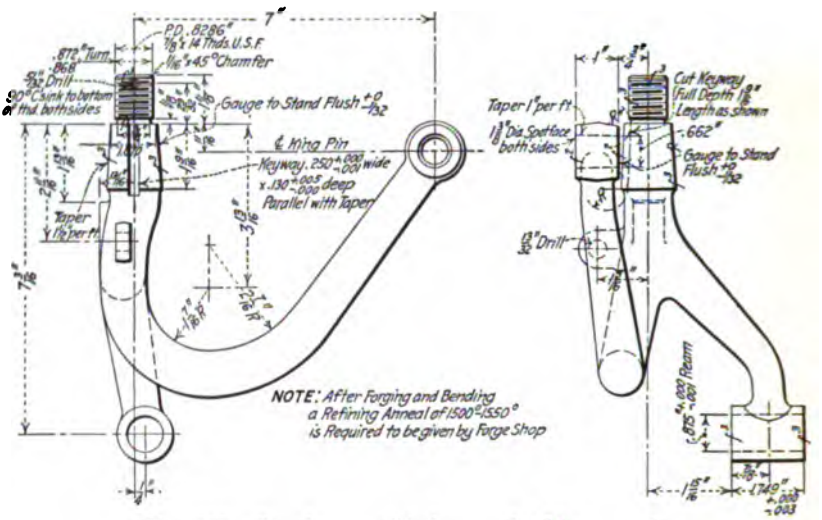

7"
P.D. .8286"
1/8"-14 Thds. U.S.F.
1/16" x 45° Chamfer
.872" Turn
.868"
Gauge to Stand Flush +0 -1/32
King Pin
Keyway .250 +.000 -.002 wide
x .130 +.005 -.000 deep
Parallel with Taper
Taper 1" per ft.
Taper 1" per ft.
Cut Keyway Full Depth 1/8" Length as shown
.662"
Gauge to Stand Flush +0 -1/32
NOTE: After Forging and Bending a Refining Anneal of 1500°-1550° is Required to be given by Forge Shop

FIG. 435.—Steering arm (Torbensen, for ¾-ton truck).

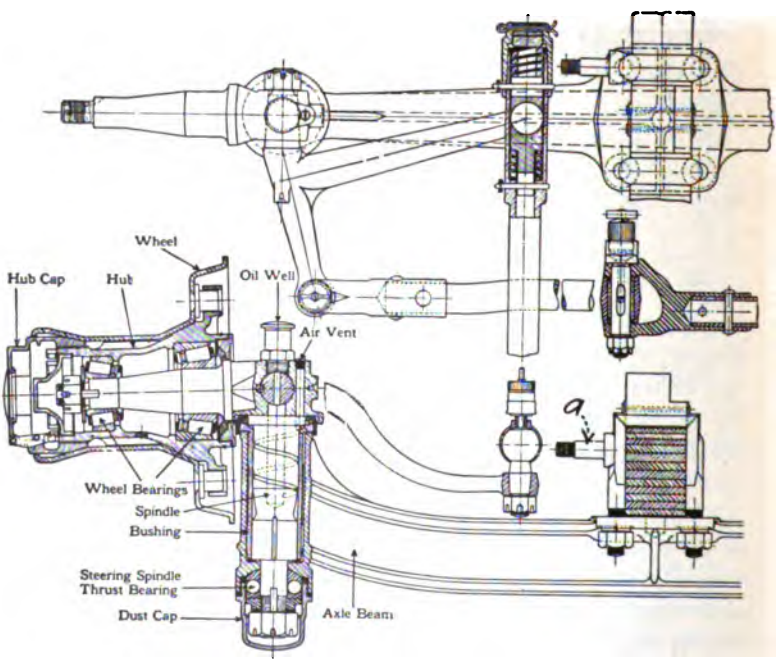


FIG. 436.—Marmon front axle.

knuckle seated directly into the axle end, the usual knuckle pin or king pin is eliminated. A cap which screws into the bottom of the steering knuckle boss eliminates the escape of oil from the bottom of the steering spindle. The space between the spindle and the bushing, at the top, is closed by a dust cap threaded to the bushing and by the projecting shoulder of the steering spindle. Lubrication is provided by means of an oil cup on the top which communicates with the reservoir in the knuckle spindle.

An air vent is provided from which the plug must be removed when filling the oil reservoir. The projection *a* at the side of the spring clip is used for attachment to one end of a shock absorber. The axle section of the Marmon is 2½ in. high by 2 in. wide with the ribs 5/16 in. in thickness. The steering knuckle is made of chrome-nickel steel, heat treated.

Figure 437 is the front axle of the Packard twelve-cylinder car (having a bore and stroke 3 by 5 in., weight of car without passengers about 4,500 lb.). In this design the knuckle pin is set at an angle of 3½° from the vertical, while the axle spindle is 2½° below the horizontal, the angle between the spindle and the axis through the knuckle pin being 84°. The turning radius of the car is 23 ft. The steering knuckle is of the inverted Elliott type. The axle head has a taper fit for accommodating the knuckle pin, the object being to tighten the pin (move it further down into the axle end) in the event of wear. Even though the pin does not oscillate, but is rigid in the axle, the great lateral strain imposed on the axle head, as for instance in skidding, will in time cause the bore in the axle end to "bell" out, and by having a tapered fit, it is easy to compensate for this wear. Other manufacturers use a straight fit and hardened bushings which are replaced when worn. In this construction the knuckle arm for the tie-rod connection is attached to the lower fork of the knuckle while the steering arm is fixed to the upper yoke.

In the Packard twelve-cylinder car, the knuckle arms and tie-rod ends are provided with ball-and-socket joints instead of the usual clevis connection. The balls are drilled for lubrication as shown in Fig. 437*a*, which is a detail of the steering ball of the Packard six-cylinder car, weighing about 3,000 lb. In this model the tie-rod is provided with the usual clevis type of end connection. The front axle is made of .40 straight carbon steel.

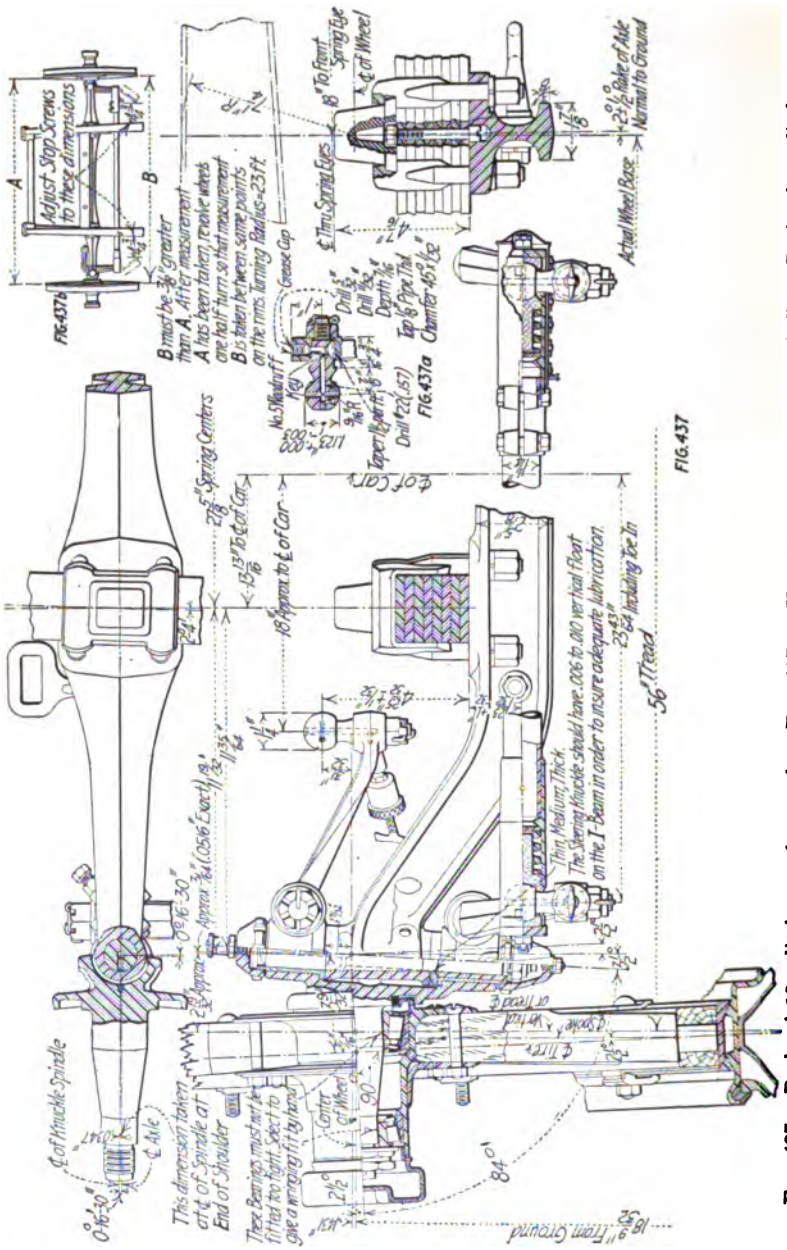

Adjust Stop Screws to these dimensions
FIG.437b
B must be ¾" greater than A. After measurement A has been taken, revolve wheels one half turn so that measurement B is taken between same points on the rims. Turning Radius=23 ft.
Grease Cup
No. 7 Taper Reamer
Drill ¹³⁄₆₄"
Drill ⅞" Deep
Tap ⅛ Pipe Thd.
Chamfer 45° Dp.
FIG.437a
¢ Thru Spring Eye
18" To Front ¢ of Wheel
2½° Plane of Axle Normal to Ground
Actual Wheel Base
278" Spring Centers
13⅞" To ¢ of Car
18" Approx. to ¢ of Car
FIG.437
¢ of Knuckle Spindle
¢ Axle
This dimension taken at ¢ of Spindle at End of Shoulder
These Bearings must not be fitted too tight. Select to give a working fit by hand
Center of Wheel
Thin Medium Thick
The Spring Knuckle should have .006 to .010 vertical float on the I Beam in order to insure adequate lubrication
56" Tread
Trim Around

FIG. 437.—Packard 12-cylinder car front axle. FIG. 437a.—Knuckle arm and steering ball for Packard 6-cylinder car.

Figure 438 is the front axle of the Mercer car (weighing approximately 4,050 lb.), having a steering knuckle pin with the top and

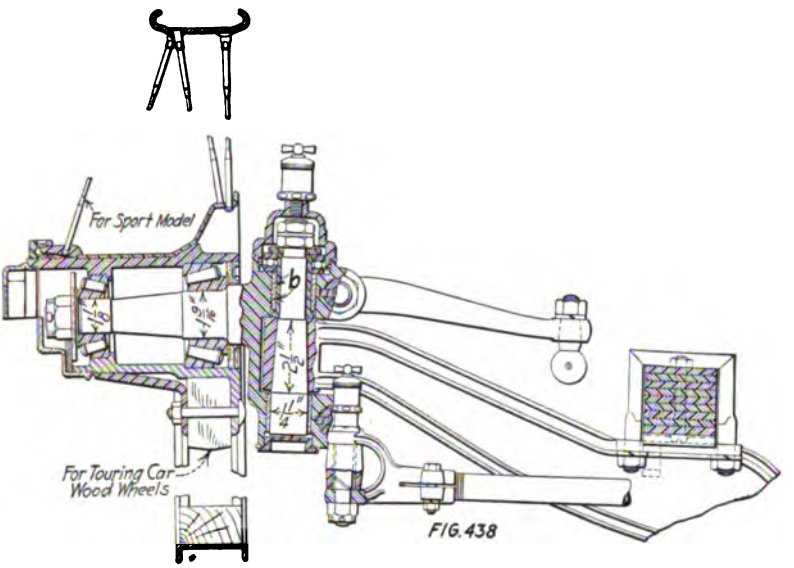


FIG. 438

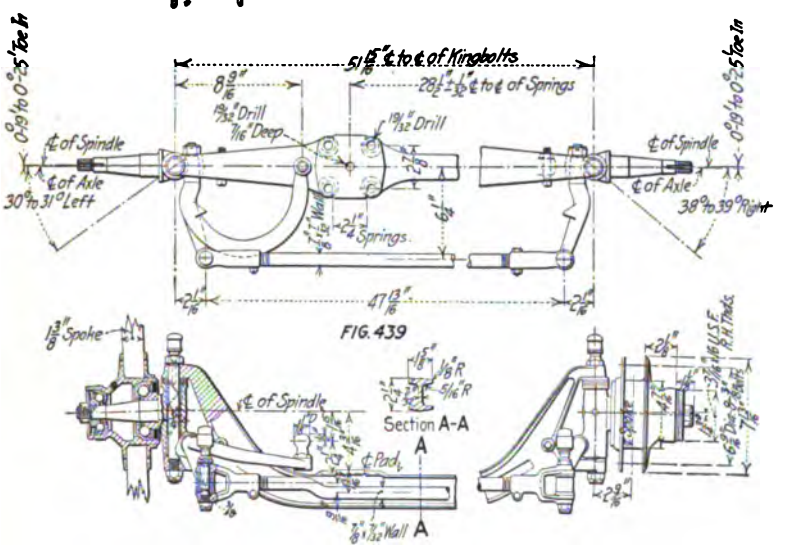


FIG. 439

FIG. 438.—Mercer car front axle. FIG. 439.—Front axle (Flint Axle Co.) of eight-cylinder Oldsmobile.

bottom bearing surfaces of different diameters, and set into the axle end from the bottom. In this construction (which is of the inverted

Elliott type) the front axle is suspended through the ball bearing and the two nuts of the knuckle pin on the top. The upper end of the pin is fitted with a steel sleeve which wears against a bronze bushing *b*, while the lower end of the pin bears directly against the bronze bushing. By using a steel sleeve on the upper end it is possible to make the two bushings interchangeable and what is more important, from a manufacturing standpoint, the

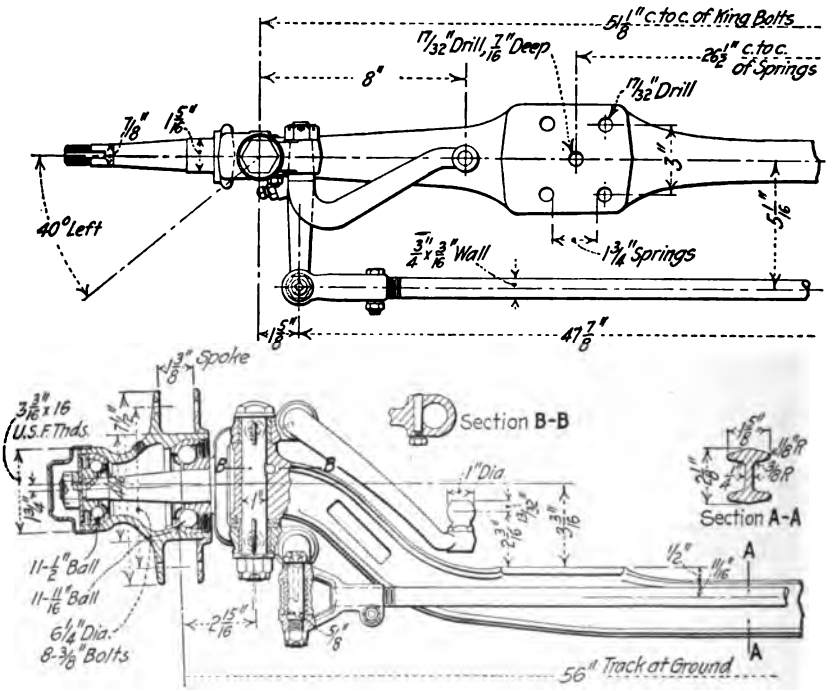


FIG. 440.—Oakland car front axle.

knuckle can be bored and line-reamed straight through. A steel dust cap is used on the top of the knuckle head, provided with a grease cup, while the lower end is closed by a cap, as shown.

Figure 439 is the front axle of the eight-cylinder Oldsmobile (made by the Flint Axle Co.). In this design the wheels are running on ball bearings. The car weighs 3,150 lb. Bronze bushings are provided in the steering knuckle.

Figure 440 is the front axle used on the Oakland car. This car has a wheel base of 115 in., a six-cylinder motor with $2\frac{13}{16}$-in.

bore by 4¾-in. stroke; weight of car without passengers about 2,450 lb. This axle is provided with the inverted Elliott type of steering knuckle but in other respects is similar to that of the Oldsmobile. The wheels are supported on ball bearings and the

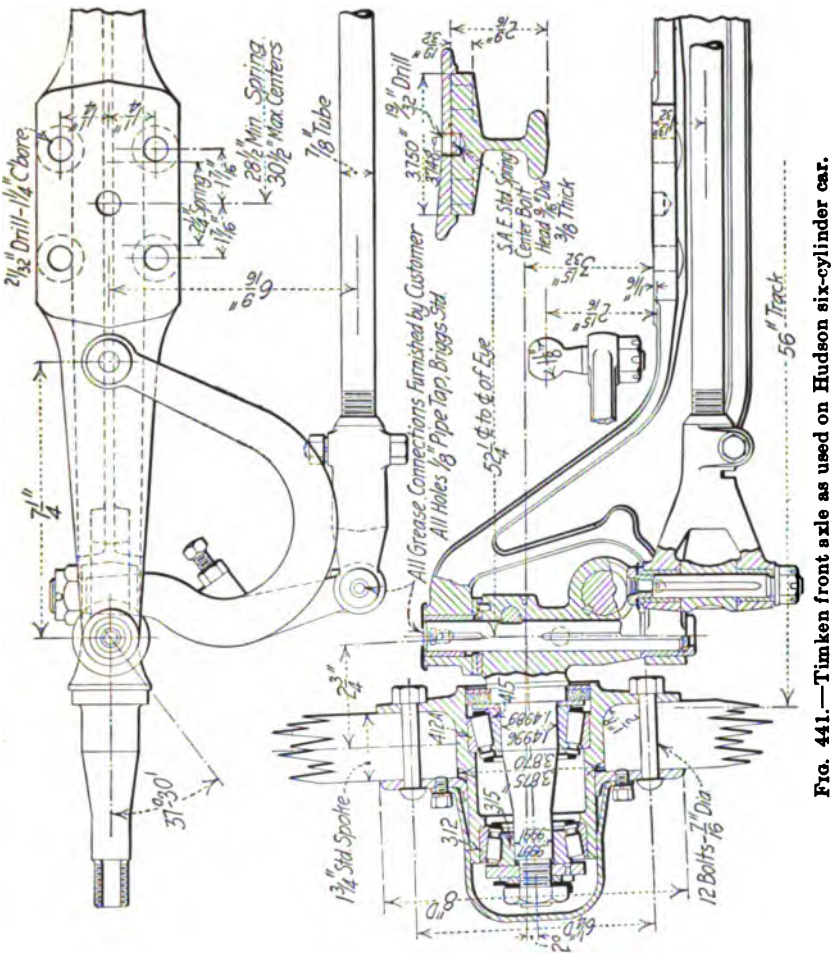


FIG. 441.—Timken front axle as used on Hudson six-cylinder car.

knuckle pin is rigidly held in the axle head. The knuckle yokes are provided with bronze bushings. As may be noted from the drawing, the knuckle pin is 1 in. in diameter, the clevis pin ⅝ in., tubular, with a wall thickness of 3/16 in.

Figure 441 is a Timken front axle suitable for a car weighing,

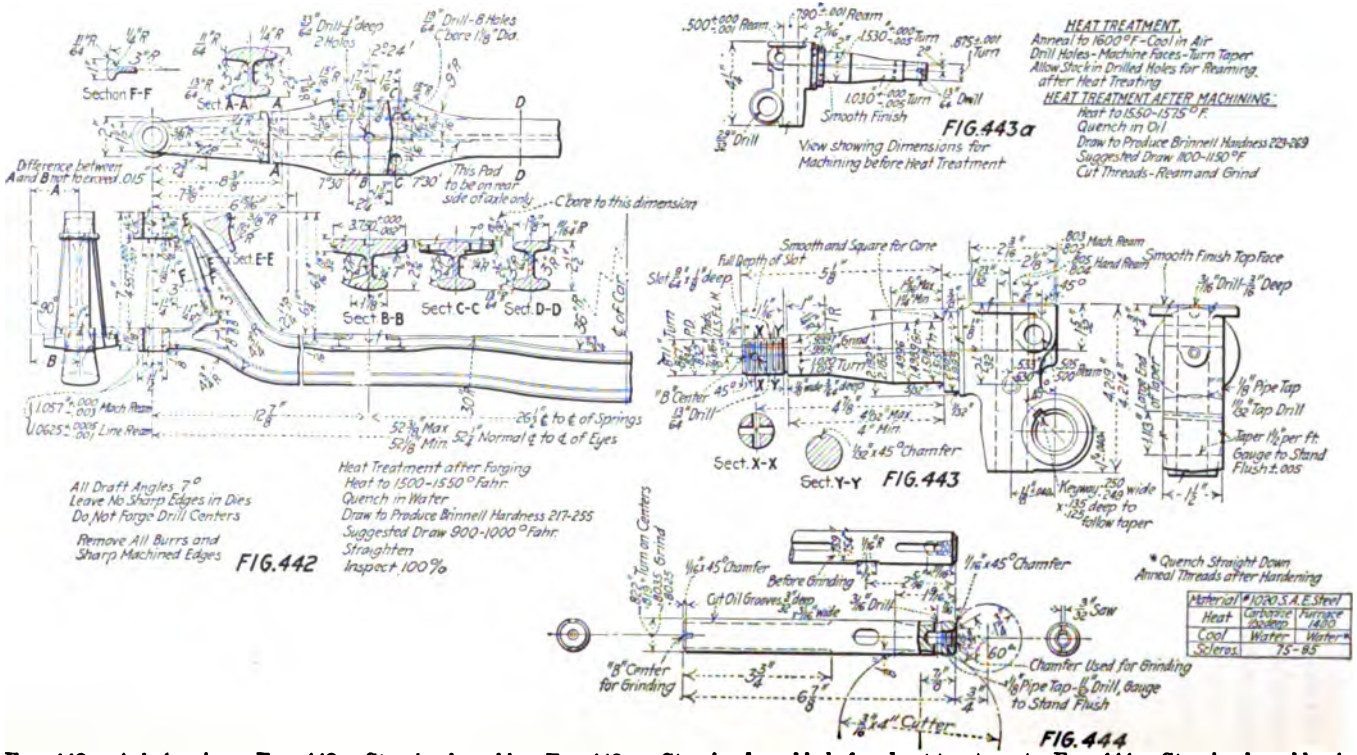


Fig. 442.—Axle forging. Fig. 443.—Steering knuckle. Fig. 443a.—Steering knuckle before heat treatment. Fig. 444.—Steering knuckle pin.

without passengers, about 3,800 lb., as used on the Hudson six-cylinder car (bore 3½ in., stroke 5 in.). In this axle the ordinary Elliott type of knuckle is used.

Figure 442 is a detail of the axle forging as used on the Hudson car. It is a drop forging of 1035 S.A.E. steel. All the details and dimensions are given and students are advised to carefully note how the sections change in order to obtain strength with lightness. All draft angles are 7°.

Figure 443 is a detail drawing of the steering knuckle, which is drop forged of chrome-nickel steel. This being a working drawing, the dimensions for both the turning and grinding dimensions are given, where necessary.

Figure 443*a* shows the dimensions for machining before heat treatment; the heat treatment is given underneath this figure, as specified by the Hudson Motor Car Co.

Figure 444 is a detail of the steering knuckle pin made of 1020 S.A.E. steel, hardened and ground. Some manufacturers mark on the drawing the machining operations in their consecutive order; in the production of this piece it is stated that the first and second operations are making the centers at the ends of the pin; the third is turning and grinding the pin on the centers; the fourth operation is drilling the oil holes; the fifth, cutting the flat as shown in the upper figure; the sixth, cutting the two oil grooves at the bottom, and the seventh, cutting the upper oil groove.

Figure 445 is a detail of the tie-rod clevis made of 1035 S.A.E. steel. The tie-rod yoke is attached to the threaded end of the tie-rod (cross-tube) and clamped tight by a clamping bolt; for this reason it is slotted as shown.

Figure 446 is a detail of the tie-rod clevis pin made of S.A.E. 1020, cold rolled steel, carbonized and hardened. The heat treatment is given on the drawing. The diameter of this pin is 6¼ in. while the cross tube yoke is reamed to .8750 in., for the hardened steel bushings, as may be noted from the assembly drawing.

Figure 447 is the Columbia front axle for a car weighing from 2,000 to 2,600 lb., without passengers. The front wheels are equipped with Bock tapered roller bearings. To limit the cramping angle of the wheels, *i.e.*, the steering angle, adjustable screws *S* are provided. Bosses, forged integral with the steering arms, strike these screw stops. The stops should be so adjusted that the tire does not approach the frame nearer than 1 in.

The steering knuckle pin or king pin is ¾ in. in diameter, hardened and ground, and turns in hardened and ground steel bushings in the axle yoke. The upper bushing is oiled through a hole drilled in the top of the king pin, the lower bushing through an oil hole drilled through the side of the axle yoke. On the lighter models the upper king pin head is provided with hardened and ground steel thrust washers while the heavier models have a

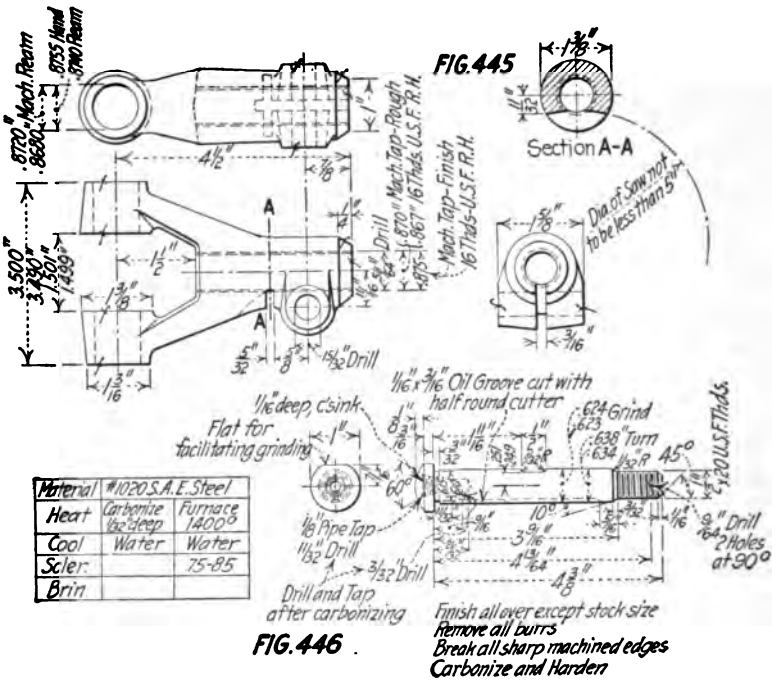


FIG. 445.—Tie rod clevis. FIG. 446.—Clevis pin.

ball thrust-bearing at this point. The thrust washers as well as the ball bearings are protected from dust by metal guards as shown, and also to retain the oil which may flow through the thrust washers or through the center of the ball bearing cups. The hardened and ground tie-rod bolts work in bronze bushings. They are lubricated by oilers on the top of the bolt, and oil holes are drilled down to communicate with the bearing surface.

The steering arm ball, which has a tapered shank, has a diameter of 1 in., while for cars weighing up to 4,000 lb., without passengers, its diameter is 1¼ in., also for commercial cars

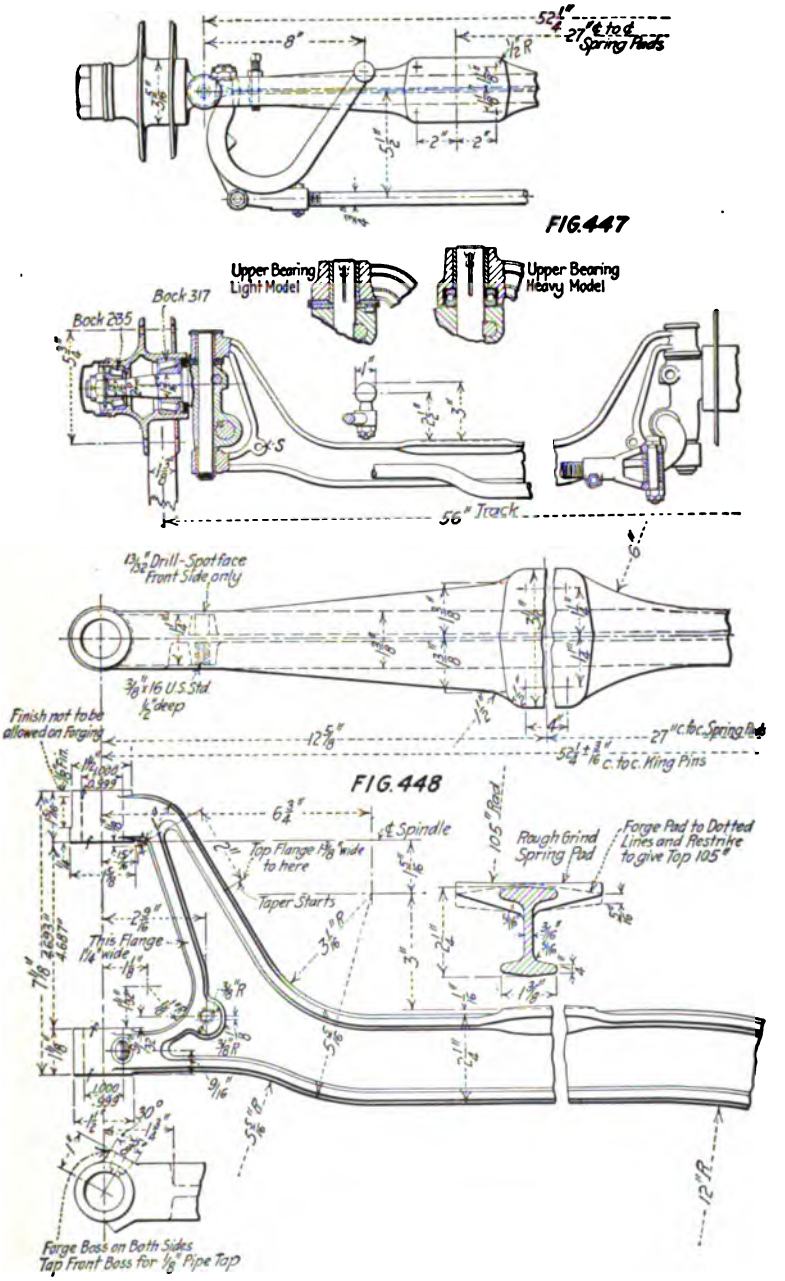
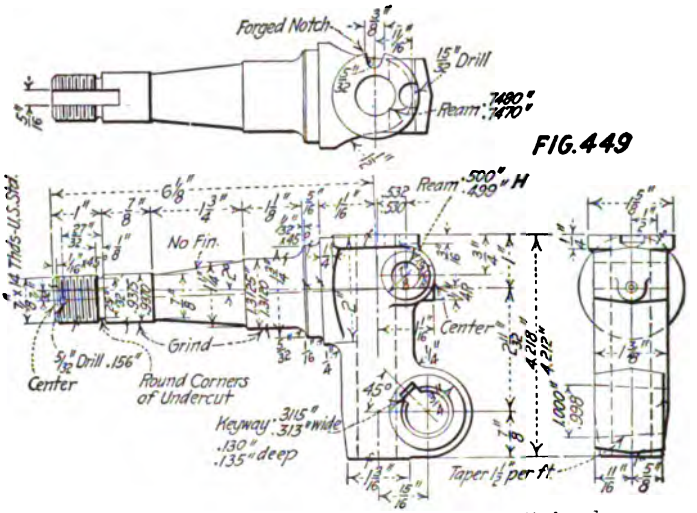


FIG. 447
Upper Bearing Light Model
Upper Bearing Heavy Model
Bock 317
Bock 235
56" Track
FIG. 448
Finish not to be allowed on Forging
Top Flange 1 1/8" wide to here
Taper Starts
This Flange 1 1/4" wide
Rough Grind Spring Pad
Forge Pad to Dotted Lines and Restrike to give Top 105°
Forge Boss on Both Sides Tap Front Boss for 1/8" Pipe Tap

Fig. 447.—Columbia front axle. Fig. 448.—Axle forging (for passenger cars weighing 2,000 to 2,600 lb.).

having a pay load capacity of from 2,000 to 3,000 lb., and 1½ in. for trucks having a capacity of from 4,000 to 5,000 lb. The



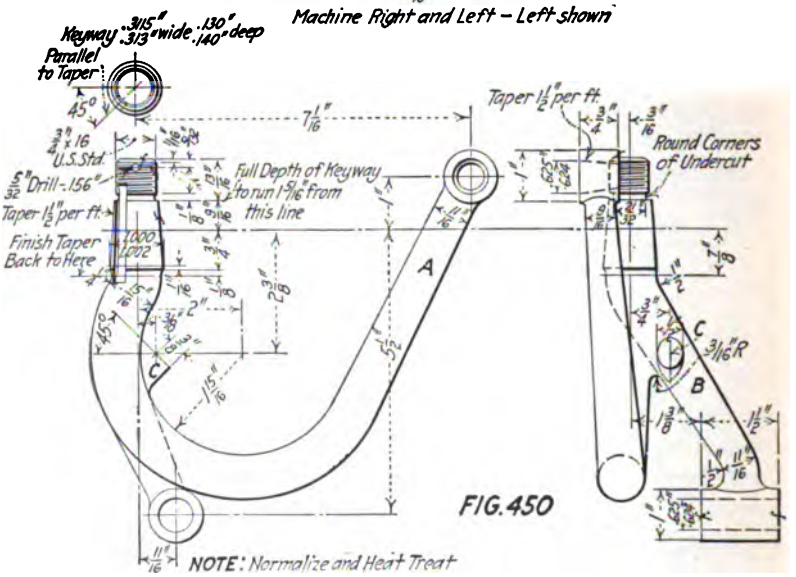


Fig. 449.—Steering knuckle. Fig. 450.—Left knuckle arm (Columbia front axle for cars weighing 2,000 to 2,600 lb.).

tie-rod ends are threaded into yokes to permit adjustment for aligning the front wheels. A slight caster effect is also given the

wheels so that the reaction from the road on the front wheels automatically tends to keep them in a straight ahead position.

Figure 448 is a detail of the axle forging with all the dimensions. Figure 449 is a detail of the steering knuckle, drop forged of S.A.E. 1035 steel. It is heat treated to have a brinell hardness of from 200 to 250. The upper hole *H*, ½ in. in diameter, serves for a bolt, flattened at one side, which prevents the knuckle pin from turning in the knuckle.

Figure 450 is a detail of the left knuckle arm *B*, forged integral with the knuckle steering arm *A*. It is made of the same material and similarly heat treated as the steering knuckle. The right hand steering arm is composed of arm *B* only. *C* is the boss previously referred to, used for limiting the cramping angle. The knuckle arms are keyed to the knuckles and have a tapered fit.

In order to check the material of important structures after they are heat treated, some manufacturers like the Cadillac Motor Car Co., who forge their knuckles of chrome-nickel steel, provide a small nipple which is forged on the yokes and later removed and examined.

The Steering Gear Drag Link.—The drag link, often called the steering gear connecting rod or reach rod, connects one of the steering knuckles with an arm of the steering post. As a rule, both ends of the drag link are provided with sockets to fit the balls of the steering knuckle arm and the arm of the steering post. To absorb vibrations and shock and to eliminate rattle due to wear, a spring is usually mounted on one side (sometimes on both) of the sockets which encompass the ball. The drag link ends should be so designed that in the event of spring failure the ball cannot come out of the drag link, although this practice is not always followed. Frequently the drag link tube is made of one piece enlarged at the ends, or else a larger end is welded to it for the purpose of accommodating the ball-and-sockets. Oil and grease cup lubrication is usually provided. Sometimes the links are made of several pieces screwed together, and pinned or clamped (see Fig. 451), as used on the White 3½-ton truck.

When the balls are assembled in the link, they are in position as seen from Fig. 452. In other words, after the ball is inserted into the enlarged hole, the end socket (and spring) is forced against it by means of the adjusting plug at the end.

Figure 453 is the drag link of the Pierce-Arrow 5-ton truck.

In this link one spring only is provided at each end, one being placed at the outside of the ball socket, and the other at the inside; in other respects both ends of the drag link are similar in construction. Occasionally, when one end of the link is pro-

FIG. 451.—White 3½-ton truck drag link.

vided with two springs, the other end may have an ordinary swivel connection, as in Fig. 454, as used at the steering column end of the White 2-ton truck.

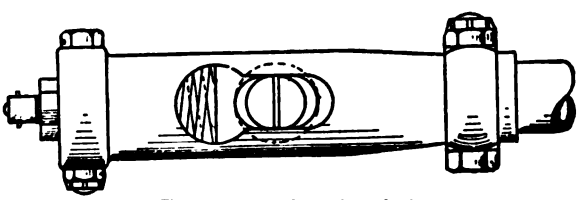


FIG. 452.—White drag link.

Figure 455[1] shows the link used on some well-known models. One spring is placed at each end of the link. The socket on the inner side is drilled and as this rod is tubular it is filled with grease, which, during operation, lubricates the wearing surfaces.

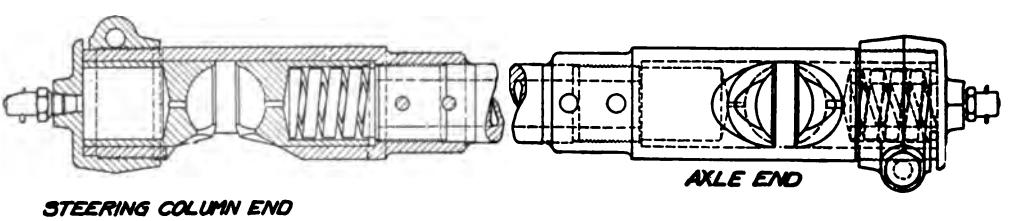


FIG. 453.—Pierce-Arrow 5-ton truck drag link.

In some constructions oil holes are provided at the sides of the tube near the socket, and wicks are placed through holes in the sockets which communicate with the oilers. By means of capillary attraction the joints are thus lubricated. To cut down the number of parts, the bottom and top ball seats are sometimes

[1] See Development of the Drag Link, by V. A. DAVISSON, *Automotive Industries*, April 22, 1920, for a number of examples.

made interchangeable, the end seat being held in place by a plug, which is pinned to the enlarged end of the tube. The last figure also shows in dotted lines how the end socket is at times made a separate piece and inserted into the plug.

Sometimes the sockets are made of bronze as in the Cadillac steering gear connecting rod, where one-half of each bronze socket

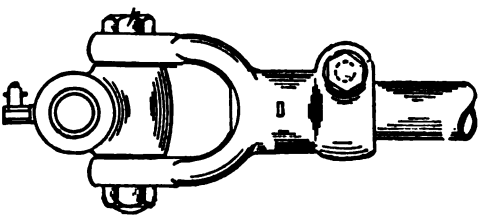


FIG. 454.—White 2-ton truck drag link (at steering column end).

is fixed and the other half is held in position by a coil spring. It is advantageous, whenever possible, to have the steering gear drag link (as well as the tie-rod) straight from end to end. Some manufacturers provide both, the reach-rod and the tie-rod, with springs to absorb shocks and vibrations and to eliminate

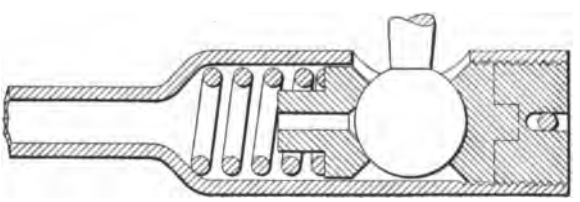


FIG. 455.—Drag link and tie rod end connection.

rattle, and the rods are made of one piece or else of several pieces, assembled or welded, as is often done. By having a plug at the end, as in Fig. 455, which extends beyond the opening of the large hole, through which the ball is inserted, it is impossible for the ball to come out.

CHAPTER XIII

STEERING GEARS

Steering Column.—The steering arm (often called steering lever or pitman arm), at the lower end of the steering column, is operated by a worm and worm wheel or sector, or by a screw-and-nut. The steering column contains the steering tube, which connects the worm or the screw with the steering wheel on the top. For passenger cars, as a rule, the ignition and throttle operating tubes or rods pass through the steering tube, which construction is termed the inside control. For trucks, ordinarily, the outside control is employed, where the spark and throttle tubes or rods are attached to the outside of the steering column. Steering gears are usually more or less irreversible, so that the motion from the front wheels, (as, for instance, when they hit ruts or obstructions on the road) may only be transmitted to the hand wheel in a small measure. If the steering gears were not made to some degree irreversible, steering of the car would be more difficult, and the motion from the front wheels would be transmitted with full force to the steering wheel and thus to the driver's hands, while if the gear ratio were such as to render the gears entirely irreversible, the strain on the front wheels, steering knuckles, knuckle arms, cross-rods, etc., would be considerably greater.

With semi-irreversible gears, the front wheels will be able to follow ruts on the road, and be deflected when hitting a glancing blow to obstructions, by transmitting the motion lightly to the steering wheel. The greater the ratio of the number of turns of the stearing wheel, or the angle through which it travels, to the entire steering angle of the front wheels, the greater is the irreversibility. For trucks, this irreversibility is made greater than for passenger cars, and the more irreversible such gears are, the greater is the necessity for springs in the drag link to absorb the impacts to some extent.

Adjustments are as a rule provided to take up the play which may develop in the gears and which cause backlash or lost

motion. Steering gears are running in a bath of oil or are packed in grease. The worm wheel is often made integral with the worm wheel shaft or trunnion shaft, to whose outer end the steering arm is attached. Thrust washers of hardened steel,

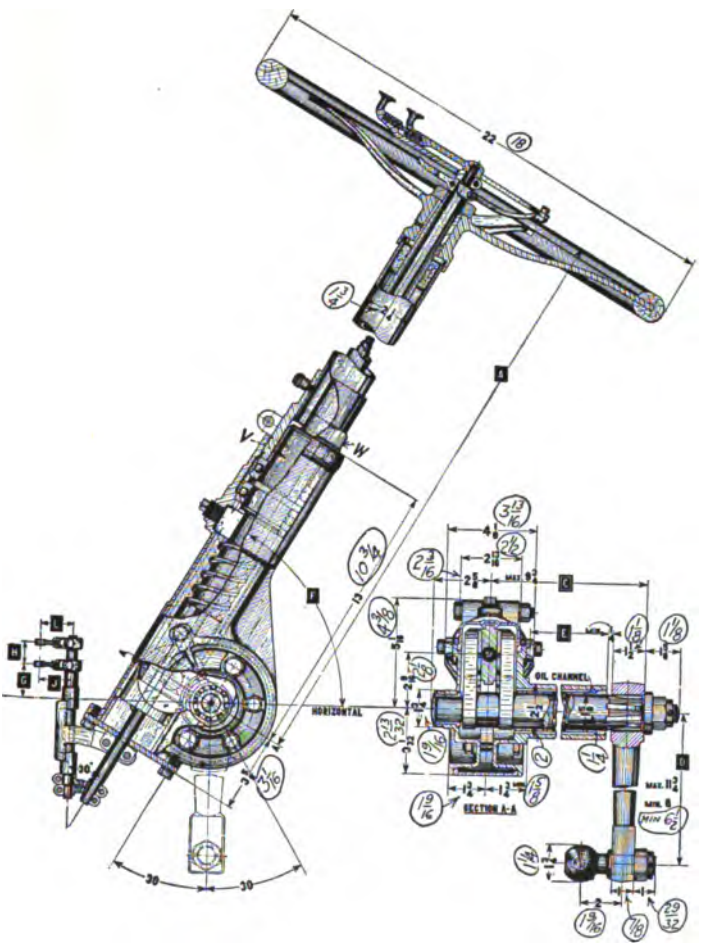


FIG. 456.—Steering gear (for from 3- to 7-ton trucks) made by Ross Gear and Tool Co.

or thrust ball bearings, are ordinarily provided to take the thrust arising from the worm as it operates the worm wheel or the sector. (When only a portion of a worm wheel is used it is called sector see part 21, Fig. 488).

The steering arm is attached to the worm wheel shaft either by

a square end or by serrations, either straight or tapered (as will be shown later); a nut at the end forces the arm rigidly upon the shaft. The lost motion should not be more than to give the wheels at their rims one inch of free motion; some manufacturers advocate a smaller amount.

Figure 456 shows the steering gear manufactured by the Ross Gear & Tool Co., suitable for trucks of from 3- to 7-ton capacity. The principal dimensions are given. The lettered dimensions from *A* to *J* are left to the choice of the truck manufacturers. The dimensions encircled are for trucks of 1½-ton capacity, or for passenger cars not exceeding 3,600 lb. When the drag-link (see page 5) extends from the back to the front, it is termed the "fore-and-aft" steering gear (which is the most common practice) and when it extends across the frame, as is sometimes the case in practice, it is called "cross-steering."

FIG. 457.—Ross Gear and Tool Co. steering gear.

The principal parts of this fore-and-aft steering gear are the screw *s* (see Fig. 457), nut *n* and trunnion shaft *t*. The screw and nut, which have very large bearing surfaces, represent the means for transforming the rotation of the steering wheel to rotation of the trunnion shaft. The steering wheel is keyed to the top of the tube *u*, to the lower end of which the screw is secured by means of a brazed joint. When the wheel turns the screw, the nut or block *n*, into which the screw is threaded, travels up or down on the screw. Longitudinal keys *k*, on the sides of the nut, engaging corresponding grooves in the main casing of the steering gear, prevent the nut or block from rotating with the

screw, therefore it can move only up or down. In the lower part of the nut are cylindrical recesses *r*, to accommodate cylindrical disks *c*, which are free to rotate in the recesses. These disks are provided with milled slots to receive the projecting arms *p*, from the trunnion shaft. Therefore, when the screw is turned, moving the nut up or down in the housing, and carrying the cylindrical disks with it, these disks oscillate and at the same time they oscillate the trunnion shaft which is journaled to the housing. When the trunnion shaft oscillates, it moves the steering arm forward or backward as the case may be.

The screw is completely enveloped by the nut in order to provide a large bearing surface between them. The threads are ground, the screw is glass hard, and the nut is heat treated, to reduce wear to a minimum. Holes are drilled in the nut, opposite every thread, in order to provide a circulation of oil through the steering gear. When the steering gear is turned hard over in one direction and the nut raised to its highest position, the oil is all forced into the lower casing. When the steering gear is turned to the other direction and the nut lowered, the oil is forced back, upward, thus displacing the oil in the bottom of the casing. This pumping action maintains lubrication in the steering gear.

The lubrication of the trunnion shaft is secured by broaching six straight oil channels or grooves (marked oil channels in Fig. 456) through the malleable iron casing, commonly known as the trunnion housing. The grooves are completely filled by projections on the outer end of the bronze bushing, between the casing and the trunnion shaft, so that the oil passing through the inside of the steering gear casing and out, through the grooves, cannot escape through the outer end. Holes are drilled in the bushings in line with the projections described above, so that the oil which passes through the main housing is forced through these grooves and the holes, thereby lubricating the bearings. In this manner adequate lubrication is insured for the trunnion shaft, as long as there is any oil in the steering gear case.

The steering arm is attached to the trunnion shaft by means of eight tapered keys milled in the end of the trunnion shaft. The sides of these keys are tapered, like the corresponding key seats in the steering arm forging. The bearing surfaces between this forging and the trunnion shaft is thus entirely on the sides of these tapered keys. A nut is used on the threaded stud to hold the steering arm in place. The advantage of this arrangement

over the taper square, is that it eliminates the tendency to open the hole in the forging as the pressure is entirely tangential and there is no spreading effort.

The threaded nut *w* (Fig. 456) is used for adjusting the thrust bearing. The jacket tube is braised into this adjusting nut, which in turn is threaded into the main housing casing. This provides a stiff column to support the steering tube and prevents the jacket tube from rotating or rattling. The tight joint between the jacket tubing and the steering gear housing prevents any grit or dirt from getting in at the top of the steering gear. The adjustment is only for taking up any wear that may develop in the thrust bearings and has no effect upon the other working parts of the gear, which do not need adjustment.

One of the features of this steering gear is that all of the working parts are steel. Figure 458 gives the details of the screw, with the shape of tooth and type of thread specified. The right hand top view shows the machining dimensions of the thread and the left the grinding dimensions. It is made of .15 to .25 carbon open hearth steel, phosphorus and sulphur not to exceed .04.

The screw is supported by two sets of ball thrust-bearings which take up the thrust in both the up and down directions. The nut or block, Fig. 459, is made from .35 to .45 carbon, open hearth machine steel, heat treated. The keys on the sides of the nut, which prevent it from turning in the housing, are ground. The cylindrical recesses at the lower end of the nut are also ground, as specified on the drawing. The cylinders, Fig. 460, which turn in the lower end of the nut are made from gray iron and are ground after being machined. Note the machining and grinding dimensions. The trunnion shaft, Fig. 461, is of high carbon chrome nickel steel, heat treated and ground all over. The steering arm, Fig. 462, is of .40 to .50 carbon, open hearth steel, heat treated, and the separate ball stud is of low carbon chrome-nickel steel, case hardened and polished. By manufacturing the steering arm from high carbon steel and heat treating it, it is claimed that a spring temper is secured which gives flexibility, durability and strength to this arm.

Figure 464 is the housing, a malleable casting containing the steering gear and Fig. 463 is the cover, one side of which supports the long bronze bushing for the trunnion shaft. The ball bearing is housed in space marked *T* while the threaded portion above it serves for the lower end of the adjusting nut *w*, shown in the assembly.

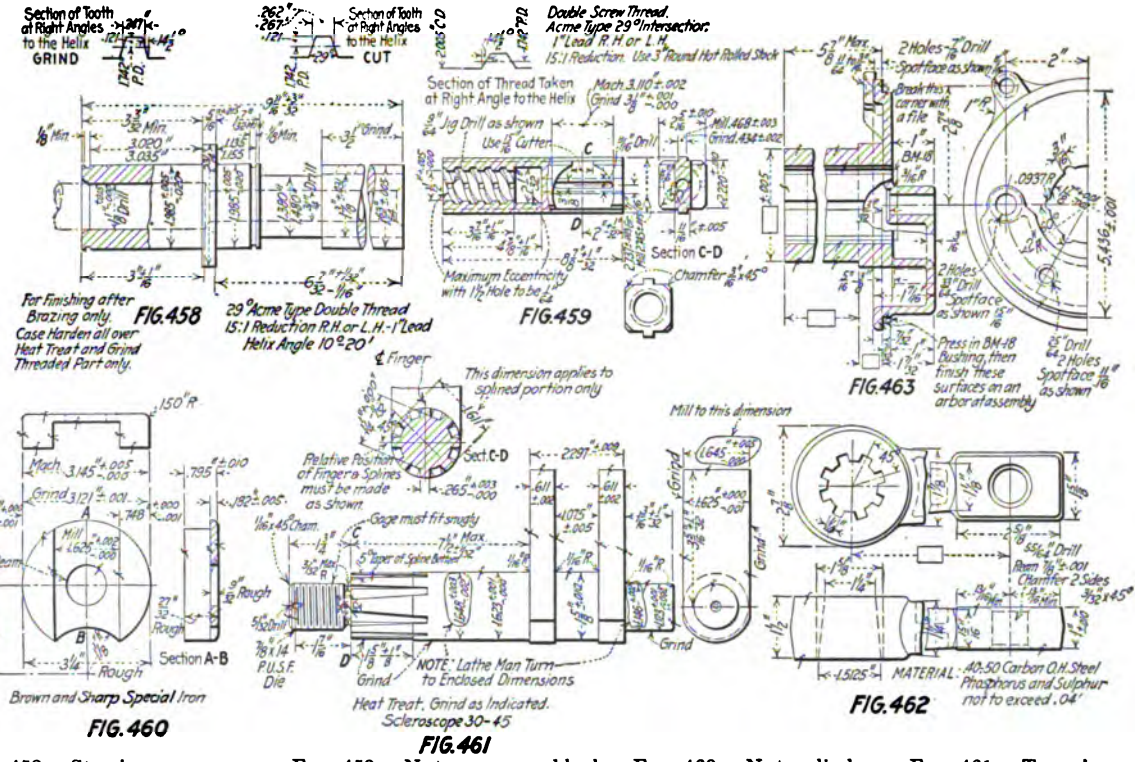

Section of Tooth at Right Angles to the Helix
GRIND
Section of Tooth at Right Angles to the Helix
CUT
Double Screw Thread. Acme Type 29° Intersector. 1" Lead R.H. or L.H. 15:1 Reduction. Use 3" Round Hot Rolled Stock
Section of Thread Taken at Right Angle to the Helix
For Finishing after Brazing only. Case Harden all over. Heat Treat and Grind Threaded Part only.
FIG. 458
29° Acme Type Double Thread 15:1 Reduction R.H. or L.H. - 1" Lead Helix Angle 10° 20'
Maximum Eccentricity with 1½ Hole to be
Section C-D
FIG. 459
2 Holes Spotface as shown
Brush this corner with a file
Press in BM-18 Bushing, then finish these surfaces on an arbor at assembly
FIG. 463
Finger
This dimension applies to splined portion only
Relative Position of Finger & Splines must be made as shown
Sect. C-D
Gage must fit snugly
Mill to this dimension
NOTE: Lathe Man Turn to Enclosed Dimensions
Heat Treat, Grind as Indicated. Scleroscope 30-45
FIG. 461
Section A-B
Brown and Sharp Special Iron
FIG. 460
MATERIAL: 40-50 Carbon O.H. Steel Phosphorus and Sulphur not to exceed .04
FIG. 462

FIG. 458.—Steering gear screw. FIG. 459.—Nut, or screw block. FIG. 460.—Nut cylinders. FIG. 461.—Trunnion shaft. FIG. 462.—Steering arm. FIG. 463.—Housing cover (Ross steering gear for 3½- to 7-ton truck).

The manufacturers build these models with inside or outside controls for spark and throttle, and with or without the small bevel gears at the bottom.

The steering gear illustrated in Fig. 456 weighs 80 lb., and is the largest size of this make. It is built in right hand and left hand models and for two reductions, 10:1 and 15:1. The former requiring one and two-third turns of the steering wheel to operate the steering arm through its travel of 60°, while with the greater reduction two and one-half turns of the steering wheel are necessary for the entire travel of the steering arm. In the smaller model built by this company, suitable for use on trucks of 1½-ton

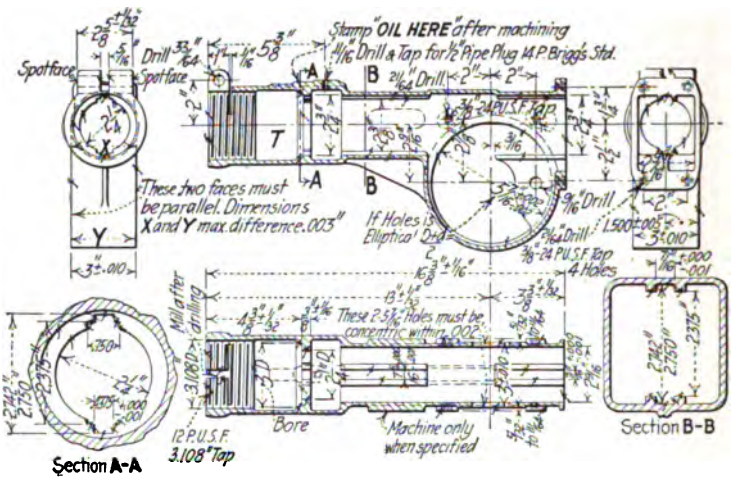


FIG. 464.—Ross steering gear housing.

capacity, where not more than 15 per cent of the pay load is carried on the front axle, or for passenger cars not exceeding 3,600 lb. in weight, the ratio is 10:1, requiring one and two-third turns of the wheel to rotate the arm through 60°. The weight of this steering gear is 40 lb. In the figures the dimensions surrounded by circles show the dimensions of corresponding parts for the smaller model.

Figure 465 is the steering gear assembly of the class B truck, previously referred to. It is the Gemmer type of steering gear, which uses a worm and worm wheel. The advantage of using a complete worm wheel instead of only a sector, as was the practice formerly, is that when wear occurs in one portion of the wheel, it may be turned to another position, say for instance, 180° from

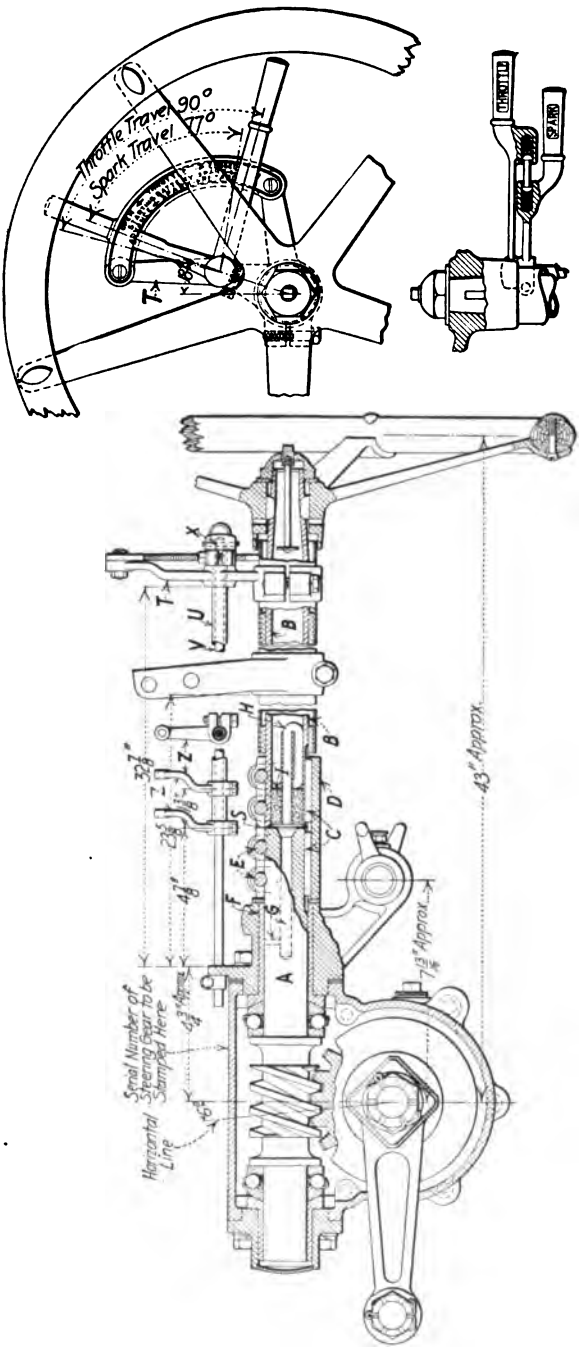
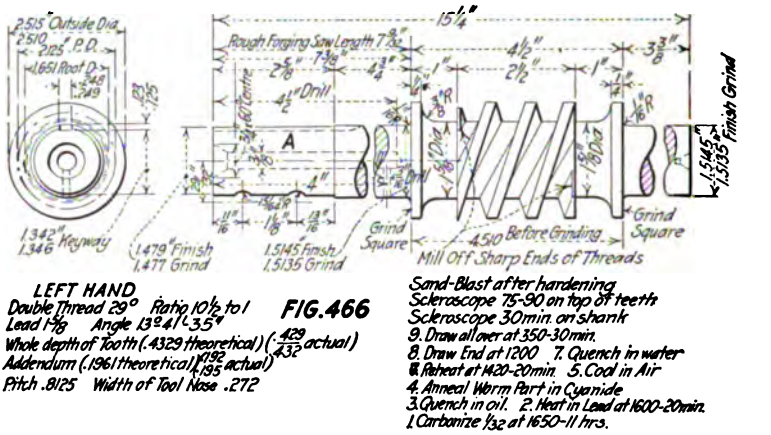


FIG. 465a.

FIGS. 465 and 465a.—U. S. A. Class B truck steering gear assembly (Gemmer type).

FIG. 465.

the original. In such event the position of the steering arm, must, of course, also be changed, in order that it assume its original position with respect to the drag link.


LEFT HAND
Double Thread 29° Ratio 10½ to 1
Lead 1⅝ Angle 13°24' 4.35"
Whole depth of Tooth (.4329 theoretical) (.429/.432 actual)
Addendum (.1961 theoretical) (.192/.195 actual)
Pitch .8125 Width of Tool Nose .272
FIG. 466
Sand-Blast after hardening
Sclerascope 75-90 on top of teeth
Sclerascope 30 min. on shank
9. Draw all over at 350-30 min.
8. Draw End at 1200 7. Quench in water
6. Reheat at 1420-20 min. 5. Cool in Air
4. Anneal Worm Part in Cyanide
3. Quench in oil. 2. Heat in Lead at 1600-20 min.
1. Carbonize 1/32 at 1650-11 hrs.

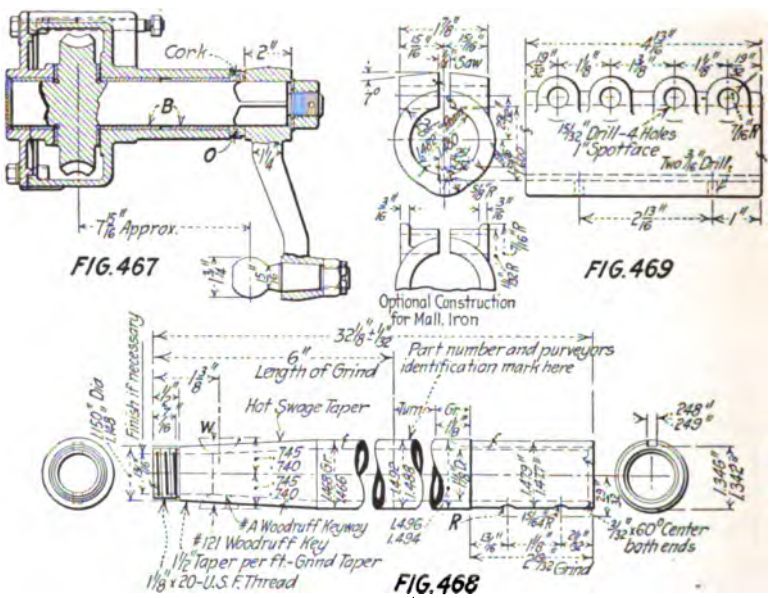


FIG. 466.—Detail of steering worm. FIG. 467.—Worm wheel assembly. FIG. 468.—Steering tube. FIG. 469.—Steering tube coupling (Class B truck).

Figure 466 is a detail of the worm, the upper end *A* being attached to the steering tube *B* by means of keys *C* and coupling *D*, in which they are located. The coupling is clamped tight by

four bolts *E*. The worm is supported at the top and bottom by ball thrust bearings to take the thrust when in operation. The worm and worm wheel are located in a practically oil tight housing which also contains the lubricant, while the upper shanks *A* of the worm run in bushings *F*, to which oil is fed through oil holes *G*, by means of wick *H* from above; *I* being a cork plug in which the metal tube, supporting the wick, is

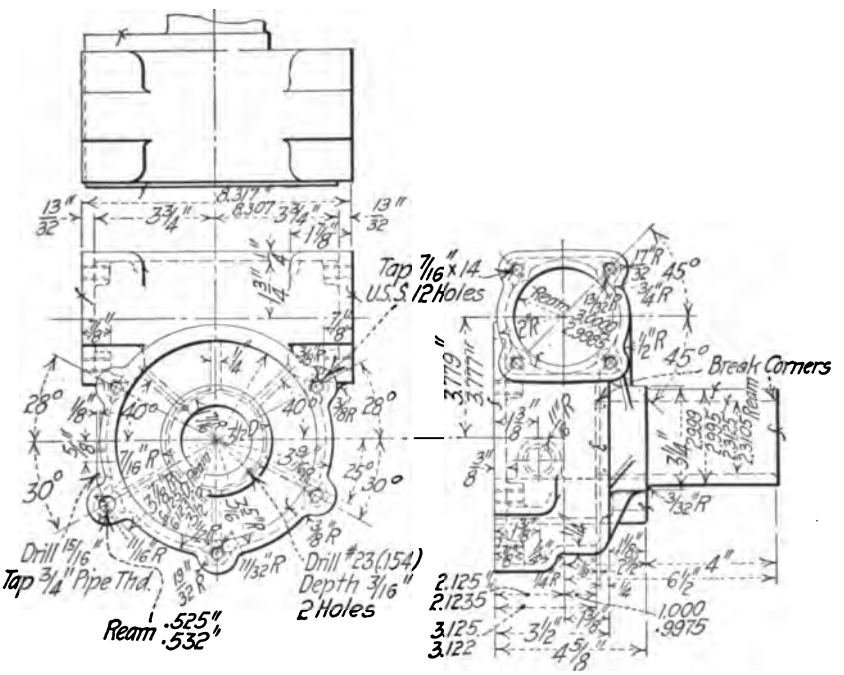


FIG. 470.—Steering gear housing (Class B truck).

located. The detail drawing of the worm gives all the dimensions and the heat treatment, the worm being made of S.A.E. 1020 carbon steel, carbonized. Special attention is drawn to the heat treatment to obtain the required scleroscope hardness on the teeth and on the shank.

Figure 467 is the steering worm wheel assembly, the worm shaft running in bronze bushings *B*. *O* represents the cork oil retainer, a small metal ring on the outside preventing the cork from spreading.

Figure 468 is a detail of the main column tube or steering tube, marked *B* on the assembly. It is made of seamless steel tub-

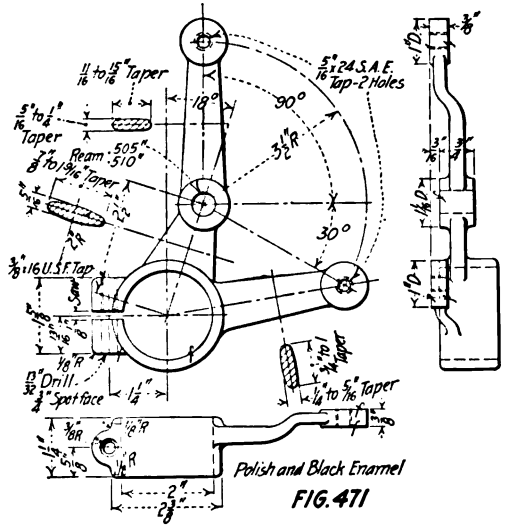


FIG. 471

FIG. 473

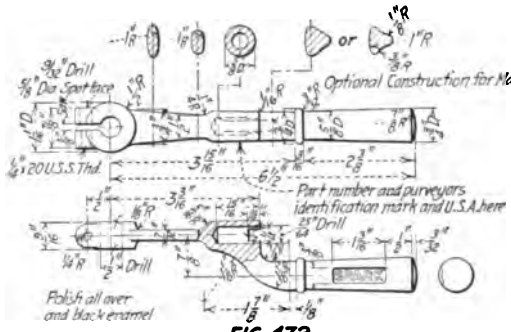

Optional Construction for Mall. Iron
1/4"-20 U.S.S. Thd.
Part number and purveyors identification mark and U.S.A. here
Drill
Polish all over and Black enamel

FIG. 472

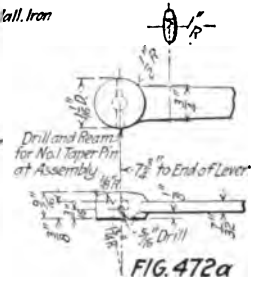


FIG. 472a

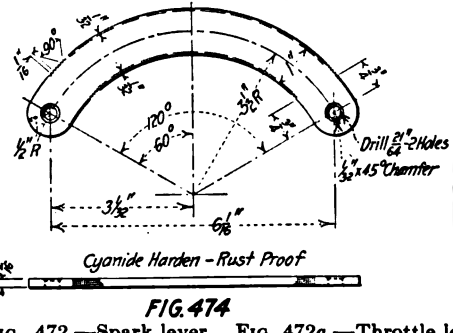


FIG. 474

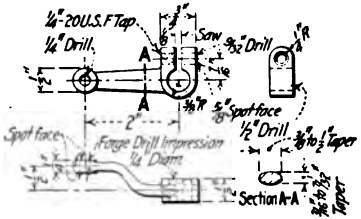


FIG. 475

FIG. 471.—Throttle and spark lever sector bracket. FIG. 472.—Spark lever. FIG. 472a.—Throttle lever. FIG. 473.—Spark control tube.
FIG. 474.—Spark and throttle sector. FIG. 475.—Lower spark lever.

ing 1½ in. outside diameter by 3/16 in. wall thickness. The two Woodruff keys *W* serve for attachment of the hand steering wheel, while the two milled out 15/64-in. radius clearances *R*, accommodate the clamping bolts of the clamp shown in Fig. 469, which is either a drop forging or a malleable iron casting; if the latter, the design is somewhat changed as shown in the lower left hand view. The wall thickness of this coupling is 5/16 in. Two separate keys *C* are employed, the dimensions of each side being .2505 to .2515. A cork washer 1/16 in. thick is used in the coupling at *S* (see assembly) to separate the steering tube from the worm.

Figure 470 is the malleable iron steering gear housing, which has a wall thickness, for the most part, of ¼ in.; it is 11/32 in. where it supports the worm wheel bearing. The sector bracket for the throttle and spark levers, which is clamped to the outside column tube, is given in Fig. 471. It is a malleable iron casting and is indicated at *T* in the assembly.

Figure 472 shows the spark lever located underneath the steering wheel, as seen from Fig. 465*a*. This lever, as well as the throttle lever, is made of S.A.E. 1020 steel. It is clamped to the upper end of the spark tube *U*, shown in Fig. 473, while the throttle lever is attached to the upper end of the throttle rod *V* to which it is pinned at *X*, (see assembly). The throttle rod is cold rolled steel 5/16 in. in diameter.

Figure 472*a* shows the end of the throttle lever, and is self-explanatory.

Figure 474 shows the spark and throttle sector which is attached to the sector bracket, previously explained. It is made of cold rolled steel 3/16 in. thick.

Figure 475 is the lower spark lever, marked *Z* in the assembly, which is either a drop forging or a malleable iron casting.

Figure 476 is the Gemmer steering gear suitable for a truck of from 1- to 1½-ton capacity and for the heaviest passenger cars. All these steering gears are ordinarily equipped with "outside control" when intended for commercial use, while for passenger cars the "inside control" is used, as shown in this figure. The ratio of the worm and worm wheel is 8½:1. In this case the worm is keyed to the steering tube by key *K*.

Figure 477 is the worm used for this steering gear. It has a left hand thread; all the dimensions are given.

Figures 478 and 479 are the details of the worm wheel and

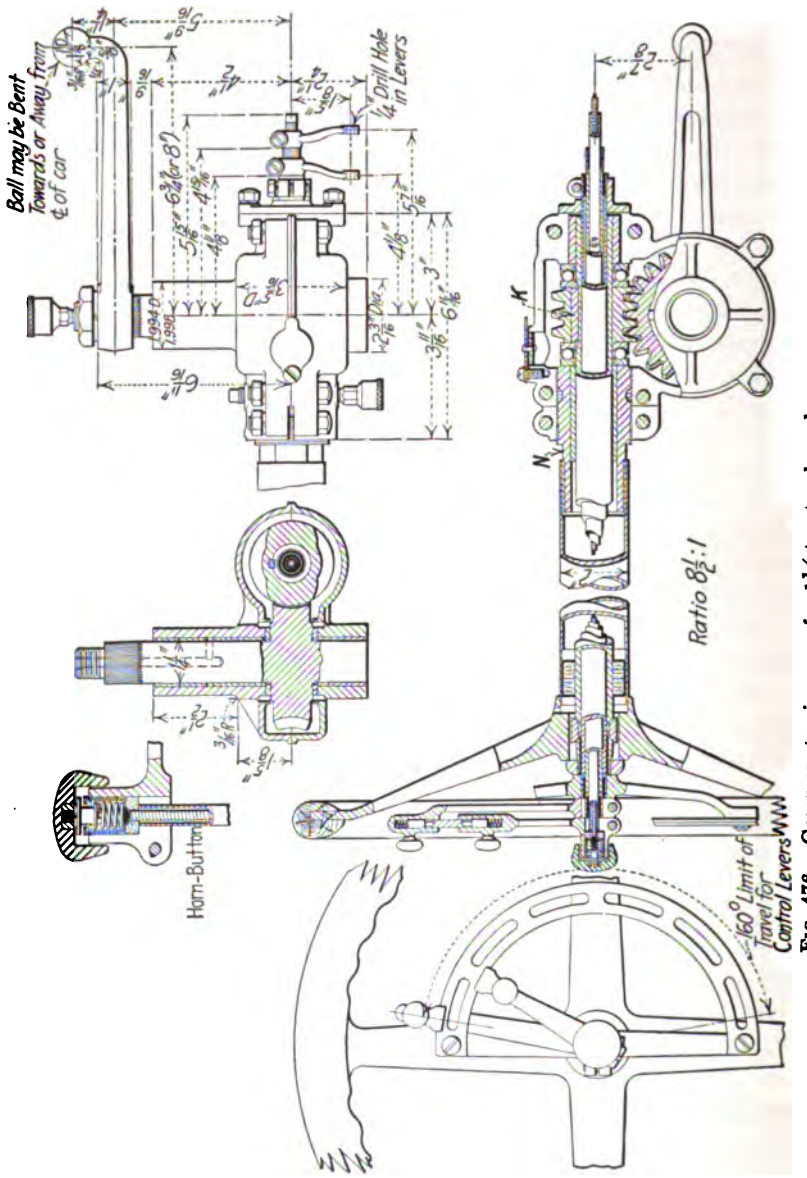

Ball may be Bent Toward's or Away from ℄ of car
¼ Drill Hole in Levers
Horn-Button
Ratio 8½:1
160° Limit of Travel for Control Levers

FIG. 476.—Gemmer steering gear for 1½-ton truck, or heavy passenger car.

shaft. It is made of 1020 S.A.E. steel, heat treated and hardened. The "insert" on Fig. 479 shows, on an enlarged scale, the serrations used at the end of the shank for attachment of the steering lever. The shank is tapered at the end where the serrations are located, a nut with lock washer holding the steering lever in place. In the lighter models, the worm wheel has a shank on only one side, see (Fig. 480), there being a screw *S* at the

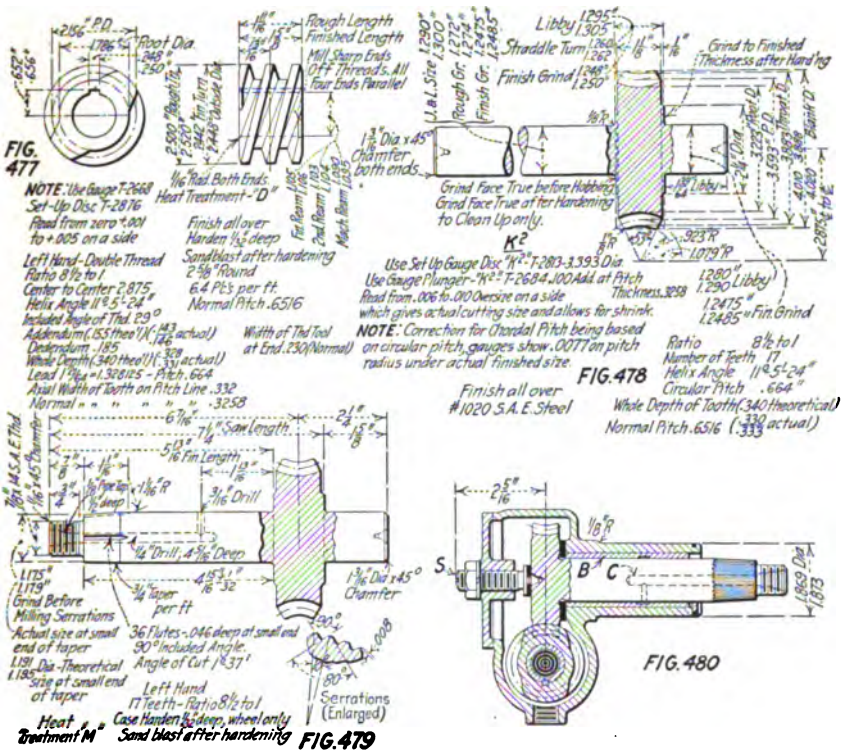


FIG. 477
FIG. 478
FIG. 479
FIG. 480
Finish all over #1020 S.A.E. Steel
Heat Treatment "M"
Sand blast after hardening
Case Harden 1/32 deep, wheel only
Left Hand 17 Teeth - Ratio 8½ to 1
36 Flutes - .046 deep at small end
90° Included Angle
Angle of Cut 18° 37'
Grind Before Milling Serrations
Serrations (Enlarged)
Grind Face True before Hobbing
Grind Face True after Hardening to Clean Up only.
Use Set Up Gauge Disc "K" T-283-3.393 Dia
Use Gauge Plunger "K" T-2684 .100 Add. at Pitch
Ratio 8½ to 1
Number of Teeth 17
Helix Angle 11° 5' 24"
Circular Pitch .664"
Whole Depth of Tooth (.340 theoretical)
Normal Pitch .6516 (.339 actual)

FIG. 477.—Steering gear worm. FIGS. 478 and 479.—Worm wheel and shaft (for 1½-ton truck and heavy passenger cars). FIG. 480.—Worm wheel for lighter cars (Gemmer steering gear).

other side, for purposes of adjustment. The shank is turning in a bronze bushing *B*, lubricated by means of oil channel *C*, shown dotted. End play in the steering post is taken out by turning adjusting nut *N* (see assembly) which adjusts the play in the ball thrust bearings. To adjust for wear in the worm and worm wheel, the bushing *B* may be made slightly eccentric, and by turning said bushing the worm wheel may be brought closer to the worm.

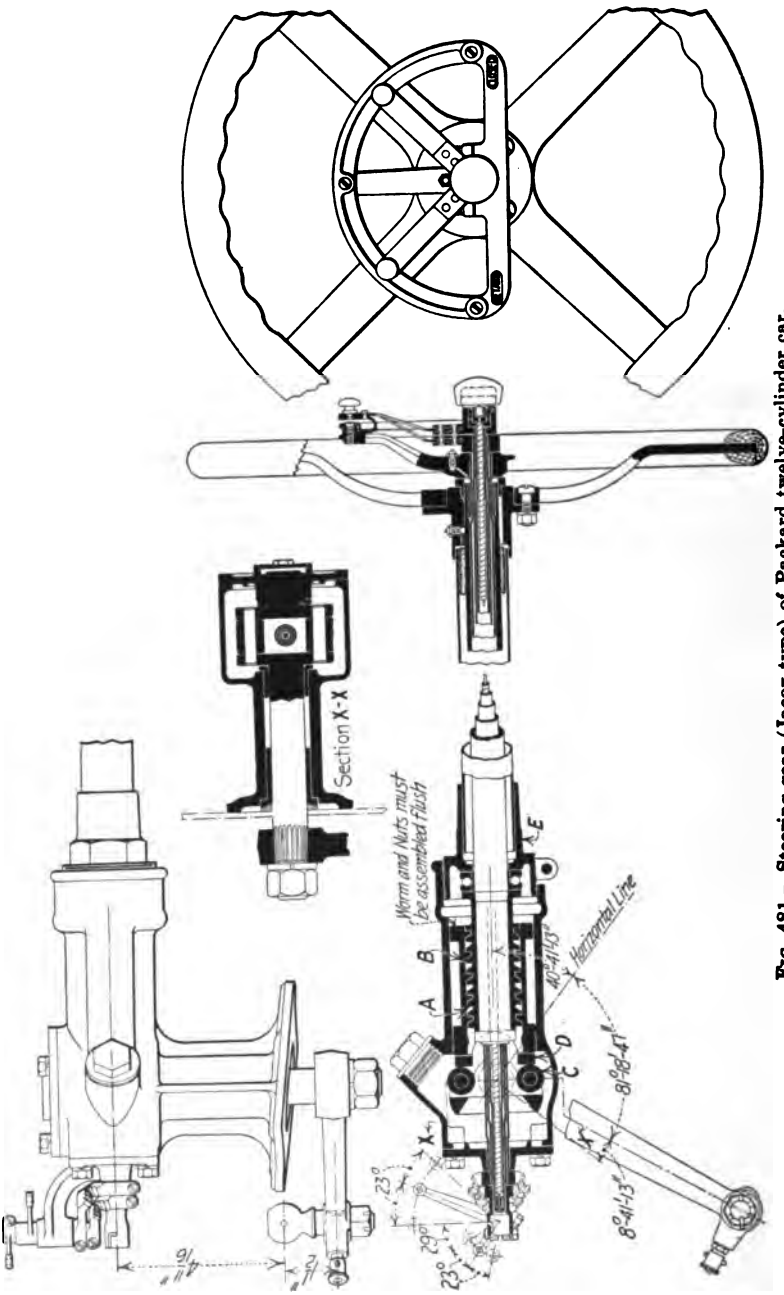


FIG. 481.—Steering gear (Jacox type) of Packard twelve-cylinder car.

Figure 481 is the steering gear used on the Packard twelve-cylinder car. This is the Jacox type of steering gear, *A* being the steering screw, *B* the right and left handed half-nuts, *i.e.*, the nut is made in two longitudinal halves, one with a right, the other with a left hand thread, as will be described later. The steering screw has a right and left handed thread which cross each other at each half turn, to mate with the "buttress" form of thread of the nut. Only one side of the threads of the half-nuts is subjected to pressure (on the side where the thread has a 3° angle from the radial plane) consequently it produces a comparatively slight radial thrust on the half-nuts. The screw is carbonized and ground on the outside diameter and at the 3° angle of the threads. The screw blanks are thread milled, carbonized, heat treated and hardened, then ground. By finishing with the grinding operation, any distortion that may have occurred by the machining or heat treatments are corrected.

In the Jacox steering gear, the half-nuts are made of bronze or semi-steel, the threads having a shearing strength of about 14,000 lb. in the pleasure car size. These gears are semi-reversible. If the steering gears are made entirely irreversible, the front wheels could not follow the ordinary ruts, but being to a certain degree reversible, the steering wheel will follow the motion of the wheels, yet a slight grip of the steering wheel will stop it from so doing. The thrust of the steering nuts is taken up by two hardened rollers *C* which are located in the same plane as the axis of the rocker shaft. These rollers roll on the flat faces of hardened thrust blocks *D* attached to the lower ends of the half-nuts. The thrust of the screw is always in one direction, for as the steering wheel rotates the screw, either one or the other of the half-nuts will be forced down, thus pressing either on one or the other of the rollers; for this reason only one adjusting screw *E* is necessary to make adjustments for wear.

The steering arm is made of 30–40 per cent carbon steel, heat treated. The bushings for the steering tube and the yoke shaft are bronze, graphite lined, so as to render them self-lubricating. It is claimed with this gear backlash can be entirely eliminated.

Figure 482 is the steering screw (made of cold rolled steel) of the Jacox steering gear, suitable for a truck of up to 2-ton capacity. The type of thread used is plainly visible; it is shown unrolled for the entire circumference, *i.e.*, four times 90°, to show the manner in which the entire screw is constructed. It is made of $1\frac{13}{16}$-in.

round stock, blanked and reamed to the dimensions shown before machining, then the outside diameter of the body is turned to 1.779–1.784. The next operation is the milling of the right and left hand thread to the root diameter shown. The screw is then carbonized $\frac{1}{32}$ in. deep, hardened and ground on the outside diameter and at the 3° angle of the thread on which the pressure is applied. (The upper side of the thread has an angle of 30°.)

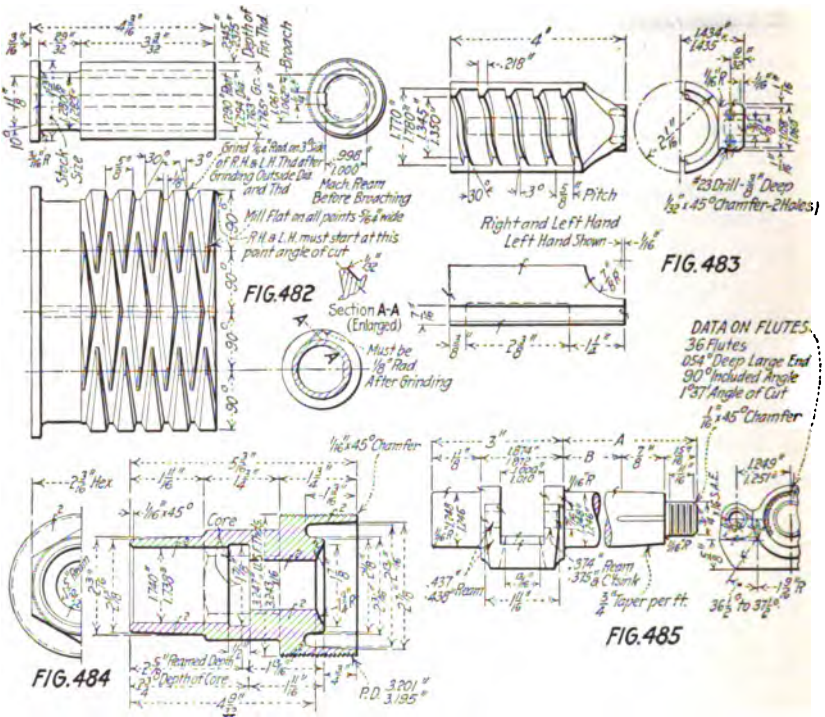


FIG. 482.—Steering screw. FIG. 483.—Left half-nut. FIG. 484.—Adjusting screw. FIG. 485.—Lever arm shaft and yoke (Jacox steering gear, up to 2-ton truck).

Figure 483 shows the left "half-nut." The nuts are cast bronze or semi-steel castings.

Figure 484 is the adjusting screw previously referred to. It is a malleable casting.

Figure 485 is a detail of the lever arm shaft and yoke, which holds the hardened rollers mentioned before. The data of the flutes or serrations are given on the drawing.

Another steering gear having a right and left hand thread on

the worm is that shown in Fig. 486, which is the Lavine gear suitable for trucks. The three principal parts of this gear are the worm shaft *a*, cut with right and left hand threads, two sliding heads *b*, which work on the worm and the trunnion shaft *c*, connecting the sliding heads with the ball arm. The parts are enclosed in housing *g*, as also seen from the assembly (Fig. 487).

The steering wheel is keyed to the upper end of the seamless tubing *d* of the worm shaft; when turning the steering wheel, the sliding heads or half nuts act on the yoke of the trunnion shaft by means of the square bushings, seen in the figures. When the steering wheel is turned in one direction, one half-nut is moving

FIG. 486.—Lavine steering gear for trucks.

down forcing one of the yoke studs down, and when turned in the other direction, the other half-nut will move down. In this manner the steering arm is moved forward or backward.

The ball dimensions are from 1⅛ to 1½ in. in diameter, depending on the capacity of the truck. The half-nuts and the worm shaft are made from bar steel .15 to .25 carbon, carbonized, hardened and ground. When wear occurs between the threads of the worm and the half-nuts, it is taken up by the adjusting screw *e* at the bottom of the worm shaft. The trunnion shaft (steering arm shaft) is a .50 carbon steel forging; it is first annealed to relieve it of internal strain, then machined and heat treated; it runs in a phosphor bronze bushing. The ball arm is attached to the trunnion shaft by means of 36 taper splines, permitting the changing of the arm into any position 10° apart. The ball arm or steering lever is a .15 carbon steel forging, annealed and heat treated. The ball pin is made of .15 to .25 carbon steel, machined, carbonized and quenched in a brine solution, the same as the sliding heads on the worm shaft, to render the wearing surfaces glass hard. Afterwards the ball

pin is polished. The gear housing g is a malleable casting, having a tensile strength of 50,000 lb. and 11.31 per cent elongation.

The Lavine gears are packed in grease and it is claimed as one

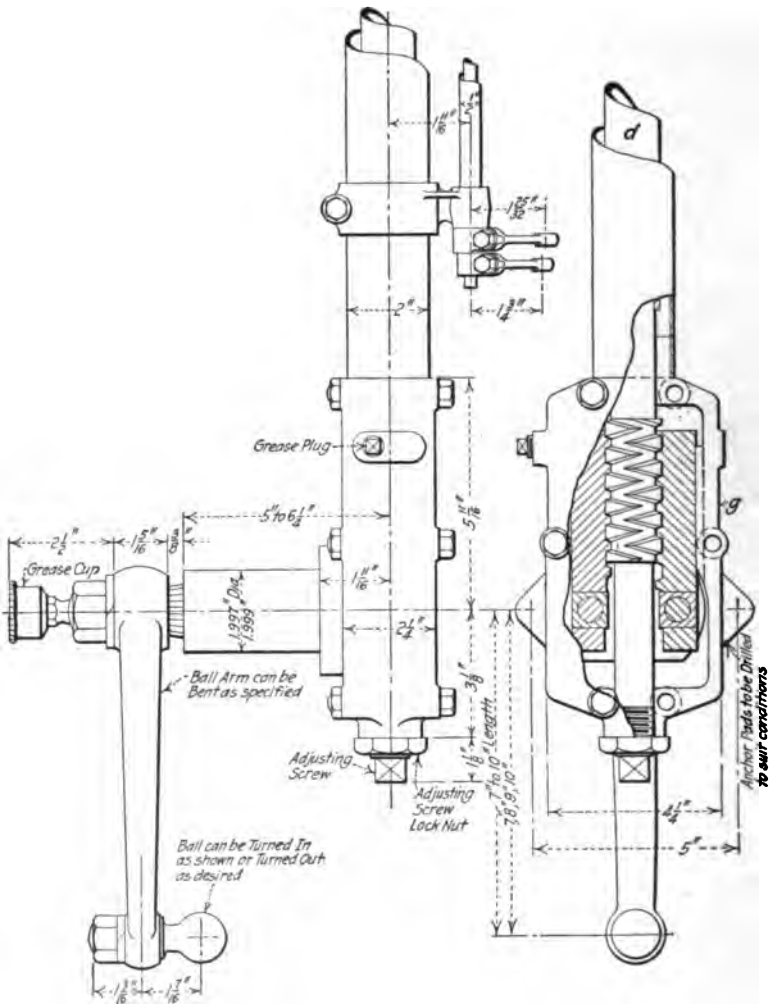


FIG. 487.—Lavine gear for trucks.

one side to the other. In adjusting this gear the manufacturers advocate to have approximately 1½-in. lost motion on the hand half-nut moves up and the other down, the grease is forced from wheel. These gears are also made with center control, in which

case the adjusting screw at the bottom is hollow to permit the spark and throttle control tubes or rods to pass through it. The steering gears for trucks having a capacity of from 1- to 2½-tons, have a reduction of 10:1, giving the steering wheel 2⅙ turns for a 70° travel of the steering arm. For the larger capacity trucks, the reduction is 15:1, and the steering wheel makes three turns for a 72° travel of the steering arm. For passenger cars the reduction is 8:1, the steering wheel making 1¾ turns for a steering arm travel of 78°.

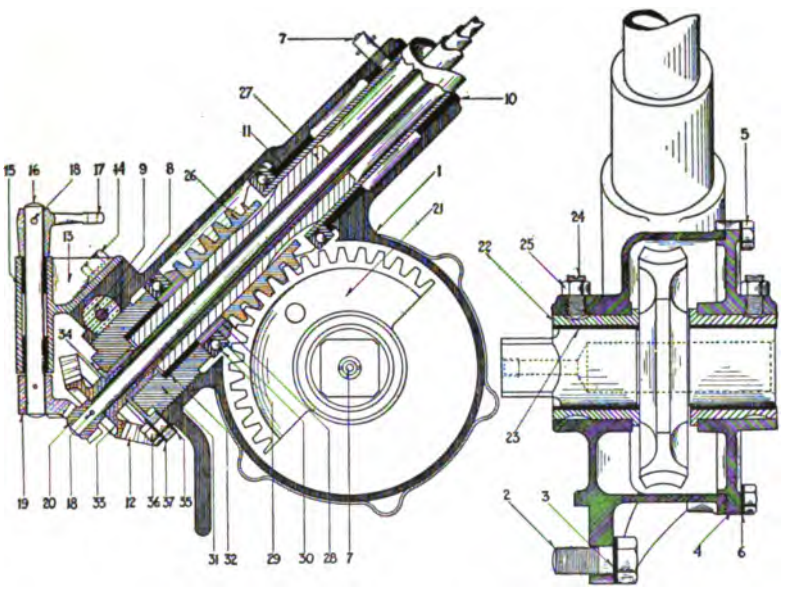


FIG. 488.—Steering gear of White trucks.

Figure 488 shows the worm and segment type of steering gear as used on the White trucks; *26* being the worm and *21* the segment. Figure *27* is the worm shaft which is supported at the bottom in the worm shaft collar *28*. Ball thrust bearings are used above and below the worm; *31* is the thrust bearing adjusting nut. Figure *32* is the thrust bearing adjusting nut bushing. Figures *10* and *11* are bushings in which the steering tube (which is connected to the worm shaft *27*) turns. As may be noted, the internal control is used; *20* and *23* are the spark and throttle control pinions.

The illustration shows the construction used on the 3½-ton truck. In the smaller models the segment is supported on one

side only, *i.e.*, it has a shaft on only one of its sides, but in other respects the construction in all the White trucks are very similar.

Steering Wheels.—Figure 489 gives the details of one of the latest designs of steering wheels, as used on the Essex passenger car. The weight of this car, empty, is about 2,600 lb.

The wheel is made of composition, with a hard rubber shell on the outside, which is provided with serrations as indicated. The center spider is made of cast aluminum with variable sec-

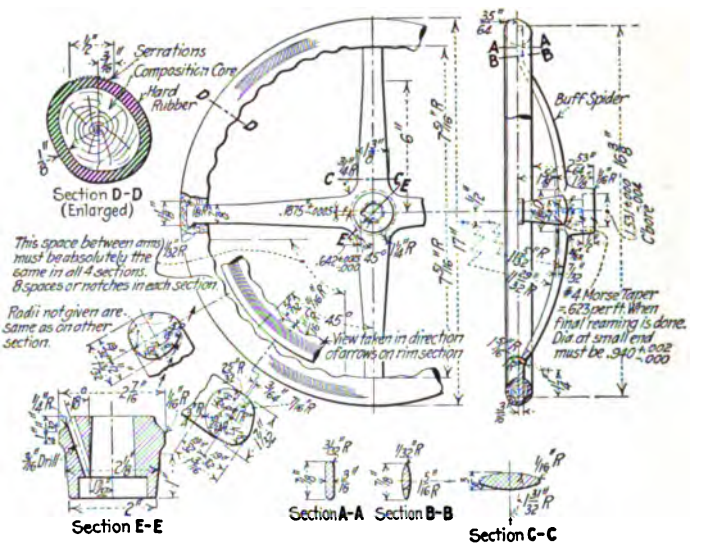

Serrations
Composition Core
Hard Rubber
Section D-D (Enlarged)
Buff Spider
This space between arms must be absolutely the same in all 4 sections. 8 spaces or notches in each section
Radii not given are same as on other section.
View taken in direction of arrows on rim section
#4 Morse Taper
Section E-E
Section A-A
Section B-B
Section C-C

FIG. 489.—Steering wheel of Essex car.

tions. The outside diameter of the wheel is 17 in. The section of the wheel itself is not round, but as shown, in order to conform better to the hand while driving. At the inner side of the wheel there are a number of notches, eight in each section, which, so to say, fit the fingers when grasping the hand-wheel. All dimensions are given.

While a number of manufacturers use compositions, or hard rubber, for the hand-wheel, others use bent wood. The diameter of the wheels for touring cars varies in the smallest cars from about 15 in., to 18 in. in the largest, while for heavy trucks the diameter is sometimes made as large as 22 in.

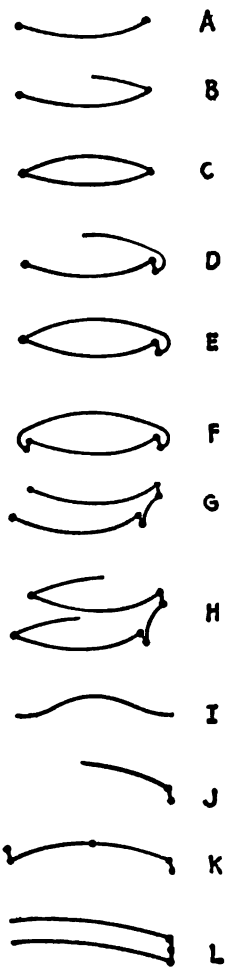
The spokes of the spider are oval, as seen from the drawing; in some designs they are of channel or *T* section, or triangular.

In some steering wheels the wood is split, the spider having a continuous ring on its periphery which is clamped between the two sections of the wheel. At other times the spiders are attached to the wheel from below by means of screws, while in the construction shown in Fig. 489, the hand-wheel is moulded with the spider in place, thus making it a one unit or integral construction.

CHAPTER XIV

SPRINGS

The various types of springs used for the suspension of motor



A—Half elliptic spring.

B—Three-quarter elliptic spring; consists of: top, quarter elliptic; bottom, half elliptic. It is joined at one end by a bolt.

C—Elliptic spring; consists of: top, half elliptic; bottom, half elliptic. It is joined at both ends by bolts.

D—Three-quarter scroll elliptic spring; consists of: top, quarter scroll; bottom, half elliptic. It is joined at one end by a shackle.

E—Scroll elliptic spring (one end); consists of: top, scroll (one end); bottom, half elliptic. It is joined at one end by a bolt and at the other by a shackle.

F—Scroll elliptic spring (both ends); consists of: top, scroll elliptic (both ends); bottom, half elliptic. It is joined at both ends by shackles.

G—Platform spring (three-point suspension); consists of: two half elliptic side members and one half elliptic transverse member. Side members and transverse member are joined by shackles.

H—Three-quarter elliptic platform spring; consists of: two three-quarter elliptic side members and one transverse member. Side members and transverse member are joined by shackles.

I—Auxiliary spring; consists of: half elliptic with plain ends.

J—One-quarter elliptic cantilever.

K—Half-elliptic cantilever.

L—Double one-quarter elliptic cantilever.

FIG. 490.—Leaf-springs.

vehicles, are given in the above spring leaf nomenclature of the Society of Automotive Engineers (Fig. 490).

S.A.E. Standard[1]

The following are spring leaf points recommended by the S.A.E. (Fig. 491).

Leaf Points[2]

No. 1	No. 2	No. 3	No. 4
Round	Half round	Square	Blunt diamond

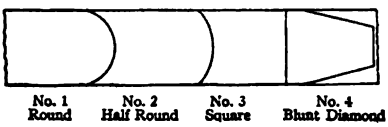


Fig. 491.—Leaf points.

S.A.E. Recommended Practice

Leaf points No. 1, 2 and 3 for rolled tapered leaves.
Leaf points No. 3 and 4 for full thickness leaves.
Rebound clips shall be used in all cases.

The half-elliptic or semi-elliptic spring is generally used for the front suspension of motor vehicles and very frequently for the rear suspension. The three-fourth elliptic *B* and full elliptic *C* are also frequently used for rear suspension, likewise the half-elliptic cantilever *K*. Formulas for determining spring deflections and dimensions are not given here, as they may be found in most engineering handbooks, but spring dimensions as recommended by the S.A.E., are given later in this chapter.

Spring Eye Bushing and Bolt Tolerances[3]

Part	Diameter tolerance, inches
Bushed eyes	+.001 −.003
Bolts for bushed eyes	−.005 −.008
Unbushed eyes	+.001 −.004
Ground bolts for unbushed eyes	−.006 −.009
Unground bolts (hot-rolled) for unbushed eyes.	−.006 Maximum, so as to use standard reamers.

[1] From the report of the Springs Division, adopted or revised by the Society, April, 1919.

[2] From the report of the Springs Division, adopted by the Society, January, 1912.

[3] From the report of the Springs Division, adopted or revised by the Society of Automotive Engineers, August, 1920.

The nominal wall thickness of spring-eye bushings shall be ⅛ in. for all sizes of bushings.

One S.A.E. "recommendation," when the drive is taken through the springs, on motor trucks, is that the second plate may be loosely wrapped around the eye of the driving end. In all other cases wrapped eyes shall not be used.

Width of Spring Ends.[1]—S.A.E. Recommended Practice. Spring ends shall be finished to a width of $\frac{1}{16}$ in. less than the nominal width of the springs, with a plus or minus tolerance of .005 in. for passenger cars and .010 in. for motor trucks, to a

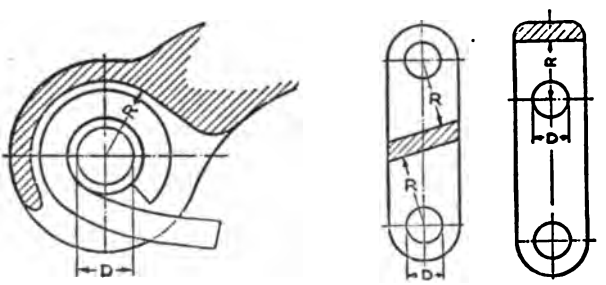


FIG. 492.—Frame brackets for springs.

point far enough back on the spring to allow free shackle movement or free sliding movement in case of flat-end springs.

Frame Brackets for Springs.—S.A.E. Recommended Practice (Fig. 492). Frame brackets at the point of connection with spring eyes shall provide sufficient clearance to allow the eye of the spring to work freely without the necessity of grinding the outer diameter of the spring eye or machining the inner radius of the hooded portion of the bracket.

The inner radius of the bracket hood shall be determined by the following formula: One-half of the bolt diameter plus $\frac{1}{8}$ in. for bushing wall, plus $\frac{3}{8}$ in. for maximum thickness of spring leaf, plus $\frac{1}{8}$ in. for clearance, or:

$$R = \frac{D}{2} + \frac{5}{8} \text{ in.}$$

where R equals inner radius of hood and D equals diameter of shackle bolt.

The distance between the ears of bracket hoods should be .010 in. greater than the finished width of the spring eyes, with a minus tolerance of .000 in. and a plus tolerance of .005

[1] From the report of the Springs Division, adopted or revised by the Society of Automotive Engineers, April, 1919.

in. for passenger cars. For motor trucks the minus tolerance should be .000 in. and the plus .010 in.

These recommendations also apply to the location of bars and the width between bosses where bar shackles are used.

The following are the spring dimensions recommended by the S.A.E.

PASSENGER CAR SPRINGS[1]

Shipping weight of car, pounds	Location	Recommended spring length, inches	Spring width, inches	Eye diameter	Spring clip diameter, inches[2]
Under 2500	Front.....	38	1¾	⅝	½
	Rear.....	52	1¾ or 2	⅝	9/16
2500 to 3000	Front.....	40	1¾ or 2	⅝	9/16
	Rear.....	54	2	⅝	9/16
3000 to 3600	Front.....	40	2	⅝	⅝
	Rear.....	56	2 or 2¼	⅝	⅝
3600 to 4200	Front.....	42	2 or 2¼	¾	⅝
	Rear.....	58	2¼ or 2½	¾	11/16

[1] From the report of the Springs Division, adopted or revised by the Society, April, 1919.

[2] Heat treated steel spring clips shall be used in all cases.

Where rear springs take drive or both drive and torque, add ⅛ in. to eye diameter on driving end. Where rear springs are used in cantilever form, the center trunnion pins should be of a diameter to equal one-half of spring width.

The table of widths and spring clip diameters is formulated not from a standpoint of theory, but from what is at the present time accepted practice, both in this country and abroad.

Shackle bolt diameters should be formulated solely from the viewpoint of possibility of lubrication, keeping the pressures at approximately 400 lb. per square inch or lower under total loads. Because of small bearing surface the bolts in the shackles should be positively prevented from oscillating in the shackles.

The term "spring clip" shall be used to indicate the forging that fastens the spring to the axle. S.A.E. Standard thread shall be used for the spring clip shank. The standard nut for spring clips should have S.A.E. hex nut dimensions, except

that the height should be one and one-half times the diameter of the threaded portion of the clip.

MOTOR TRUCK SPRINGS[1]

Nom. capacity, tons	Location	Spring width, inches	Min. length of spring, inches	Eye diameter	Spring clip diameter (alloy steel)	
					Spring set on axle	Spring underslung
¾	Front	2 or 2¼	40	¾	⅝	¾
	Rear	2 or 2¼	50	⅞	⅝	¾
1	Front	2 or 2¼	40	¾	⅝	¾
	Rear	2¼ or 2½	50	1	¾	⅞
1½	Front	2¼ or 2½	42	¾	¾	⅞
	Rear	2½ or 3	54	1	⅞	1
2	Front	2¼ or 2½	42	¾	¾	⅞
	Rear	3	54	1⅛	⅞	1
3½	Front	2½ or 3	44	⅞	⅞	1
	Rear	3 or 4	56	1¼	1	1⅛
5	Front	3	44	⅞	⅞	1
	Rear	4 or 5	56	1¼ or flat end	1	1⅛
7	Front	3	46	1	1	1⅛
	Rear	5	58	1¼ or flat end	1⅛	1¼

[1] From the report of the Springs Division, adopted or revised by the Society, April, 1919.

The column "Capacity, tons," is intended only to indicate the general truck capacities on which the corresponding usual spring sizes are used. The above table applies to all types of drives.

Where rear springs are underslung, add ⅛ in. to clip diameter. In case the rear springs take drive or both drive and torque, the spring eye should be the same in diameter on the drive end as on the other end, and in agreement with the above table. Considering strength of the main plate, it is believed more desirable to put a heavier load on a bushing than to risk opening up the eye, which might happen if the eye were increased in diameter.

Heat-treated alloy steel spring clips shall be used in all cases.

Offset Springs.—If the center-bolts are offset, the amount of offset (from the centers) should be 1, 1½, 2, or 2½ in.; for springs 36 in. and 38 in. in length, it should not be more than 1½ in., while for spring lengths of from 48 in. to 60 in. it may be as high as 2½ in. (the latter will render one end 5 in. longer than the other end). Offsets of less than 1 in. should not be used as they cause confusion in assembling.

Front Springs.—The front end of the front spring (semi-elliptic) is as a rule pivoted directly to the front end of the frame (see Brackets for front of frame, Chap. III, pages 60 and 66) while the rear end of the front spring is connected to the frame by means of a spring shackle.

When the spring is deflected, the distance from its ends to the center will vary, hence the axle will not stay in the same position with respect to the frame and one end of the spring at least must be shackled to allow for the variation in spring length. There is an advantage in anchoring the spring at the front, for in such case, when the wheel hits an obstruction and the spring is flexed (it becomes more straight) the axle will move slightly backward, and this reduces the intensity of the shock or the impact, as will be mentioned later in this chapter, under "Spring Suspension." If the front end of the spring is higher than the rear end, it will also permit the axle to move back when the spring is flexed, and this is often done in practice.

The front spring (as well as the rear) is frequently offset, as was stated before. One advantage of this construction is that it permits an increase in the wheelbase with a given frame. It also strengthens the spring as the drive through the rear spring (when no thrust rod is used) and the pull through the front spring, is taken through the shorter half of the offset spring; the car being propelled through the rear axle, while the front axle is pulled by the front half of the front spring.

The "drive" (or thrust) from the rear wheel is transmitted to the frame either through a thrust rod or through the spring itself. The torque reaction (*i.e.*, the tendency of the axle housing to rotate backward when the car is propelled forward, and *vice versa* when backing the car) is taken up by a torque rod or by the spring itself. When the spring takes both the "drive" and the "torque," it is called the "Hotchkiss drive," and this practice is largely followed at the present day.

Hotchkiss Drive.—In the Hotchkiss drive, torque, thrust and

radius rods are eliminated and the entire drive is taken through the spring. When using the Hotchkiss drive the spring camber should not be too high, and the spring seats must be fixedly attached to the axle housing, usually welded to it, and not swiveled as is necessary when a torque rod is employed. With the Hotchkiss drive, therefore, oscillating motion between spring seat and rear axle is eliminated, and no lubrication is here required. The lubrication of these parts (when a torque arm is used) is not frequently given attention by the user and they are ordinarily exposed to dirt and water. It is also claimed that with the Hotchkiss drive a smoother suspension is obtained, for when starting and stopping the action is more flexible, since the drive is taken through the spring leaves.

Opponents to the Hotchkiss drive claim that it imparts undue stresses to the spring leaves and therefore the spring has to be made stiffer. When the drive is taken through the springs the front end of the rear spring has to be pivoted to the frame without a shackle (the rear end being shackled; this refers to semi-elliptic springs). When springs are simply used to transmit the torque, both ends of the spring may be shackled, as shown in Fig. 394.

Spring Deflection.—The front spring is usually made shorter than the rear spring in order to have a considerably smaller deflection in the front than in the rear, thereby obtaining a different periodicity (vibrations per minute) than in the rear. By this method the riding quality of the whole car is improved. On the other hand, if the front spring is made too stiff, it will not be as comfortable, but for high speed cars it has been found desirable to make the front spring comparatively stiffer than for ordinary touring cars.

In the design of ¾-elliptic and platform springs, it is claimed they should be so dimensioned that the deflection of the front portion of the spring, *i.e.*, the portion projecting in front of the rear axle, should be equal to the deflection of the combined rear portions. In the ¾-elliptic, this would be the rear half of the semi-elliptic and the ¼-elliptic above it. If this is not done, some portions of the spring are liable to be over stressed and it will also tend to produce a rocking motion at the spring seat. A spring is most satisfactory in action when it has no camber, *i.e.*, when it is straight, under full load (this refers more especially to the semi-elliptic and the cantilever spring). In such

event the leaves will be subjected to purely flexure stresses, *i.e.*, the force arising from the load is exerted at right angles to the spring leaves. A wider spring is also more satisfactory than a narrower spring. A number of designers dimension the spring so that the thickest portion is no greater than its width. Springs are not only subjected to the load stresses but they are often required to take the driving stresses due to torque and torque reaction, as previously mentioned. When applying the brake the stresses in the spring, when there are no torque rods, are enormous. When the spring is used without a torque or thrust rod, or without both, it has to be made stiffer to work satisfactorily, but the loads on the spring are then also greater.

Sometimes only the torque is taken through the spring while the drive is taken through thrust rods. Figure 394 shows the worm driven rear axle of the Kelly-Springfield truck, where both ends of the spring are shackled, while the drive is transmitted through thrust rods which, at the same time, act as radius rods or distance rods.

In this construction the spring takes the torque reaction on forward drive and when applying the brake. The spring is rigidly attached to the axle housing, hence the front half of the rear spring will take the torque reaction on forward drive, since the axle will tend to turn clockwise. When the brake is applied, or when the wheels are locked, the tendency of the axle will be to turn counter-clockwise, therefore, the rear half of the spring will take most of the torque.

Cantilever Spring.—In the cantilever spring, the center of the spring is pivotally attached to the frame, while the front end is attached to the frame through a link to permit its elongation; the rear end is attached fixedly when the Hotchkiss drive is employed, and this type of spring lends itself very readily for this drive. When torque tubes are employed in connection with cantilever springs, the spring should not be pivoted to a bracket fixedly attached to the axle, for when one spring only (the spring on one side of the car) is deflected, the spring and the torque tube are stressed severely, and a spring breakage or a damaged torque tube may result.

Whenever a torque tube is used, which does not permit lateral give to the axle, it is advisable to attach the rear end of the cantilever spring to a bracket of the axle which swivels or floats thereon, so that the axle may remain in its position at right angles

to the frame, even though one spring should be deflected more than the other. Of course, if the front end of the torque tube is arranged so as to permit lateral movement, this precaution is not necessary.

Spring Periodicity.—When the number of vibrations of the rear spring is less than 95, with the car fully loaded, it will usually be found to possess sufficient comfort for the occupants. On the other hand, when the periodicity is above 100 per minute, the suspension will be found choppy and uncomfortable.

Under "Fundamentals of Periodicity" appearing later in this chapter, this matter is treated more fully, but the periodicity, *i.e.*, the number of vibrations (one oscillation up and one down) may be found from the equation: $t = 2\pi\sqrt{\frac{d}{g}}$, where t is the time in seconds of one complete vibration; d the deflection of the spring under static load in feet (*i.e.*, the difference of the spring camber, in feet, between its original camber and the camber under full load); and g is the acceleration due to gravity = 32.2 ft. per second, per second.

In order to find the number of vibrations per minute, 60 (the number of seconds per minute) is divided by the above equation. In other words, if N represents the number of complete vibrations per minute, $N = \frac{60}{t} = \frac{60}{2\pi\sqrt{\frac{d}{32.2}}}$. For example, if the static deflection is 4 in. (= .333 ft.) the periodicity is:

$$N = \frac{60}{2 \times 3.1416 \sqrt{\frac{.333}{32.2}}} = 94 \text{ vibrations per minute.}$$

It might be mentioned here that the front springs, being comparatively stiffer, have a higher periodicity. If the periodicity of both the front and the rear springs were approximately the same, the car would tend to pitch and rock. Front springs on well designed cars have a periodicity varying roughly between 125 to 155 vibrations per minute. The greater the load on the spring the lower is the periodicity, as will be shown later.

Spring Suspensions.[1]—The chief factors affecting the riding quality of a motor vehicle are: Spring deflection, or amplitude;

[1] From a paper by the author read before the S.A.E., January, 1920.

periodicity, or the number of vibrations[1] per second; and the proportion of the sprung to the unsprung weight. Other factors influencing riding quality are the wheelbase, the tread, the height of the center of gravity of the car and the effect of the front springs on the rear ones. Speed of travel naturally has an effect upon the spring suspension, and therefore on the riding qualities. We will not here consider the effect of speed or speed variation, but shall bear in mind their relative influence at all speeds, with a wheel of a given diameter and the same quality of tire.

The spring having the largest amplitude of deflection for a given load will flex most for small or large additional upward thrusts of the axle, when the wheel encounters obstructions on the road, with the least perceptible disturbance to the car body. For instance, if one spring deflects 1 in. for each additional load of 150 lb. and another moves the same distance for a 200-lb. load, with deflections not exceeding the elastic limit of the material, the 150-lb. or more flexible spring will produce the lesser intensity of upward shock, even though the time interval during which the upward thrust acts is longer.

Vibrations Per Minute.—The periodicity, as well as the amplitude of the deflection, depends on, not only the length, thickness and width of the spring-leaf, considering a single-leaf spring for the present, but also the load that it supports. The greater the static load and the greater the static deflection of the spring, or the position of rest when the load is applied, the slower the vibrations per minute.

It is not so much the height through which the body is raised as the rate of speed at which it is raised that affects the passenger's comfort, although both are of great importance. The final requirement of a good spring suspension is that it imparts to the car body the least amount of upward motion and at the slowest speed. Tests[2] have disclosed the fact that the axle, when the wheel travels over an obstruction, causes the wheel to jump over the obstacle in each case and come to the ground beyond it (see Fig. 493); and it appears that the shape of the obstacle makes no appreciable difference, unless its slope is very gentle. It has also been shown that the wheel reaches the top of its upward

[1] The term "vibrations" is used instead of "oscillations," since the best writers in physics use "oscillation" to mean one-half of a complete vibration; thus, a vibration means two oscillations, one up and one down.

[2] *Transactions* of the Institution of Automobile Engineers, vol. 7, page 451.

motion almost before the car body begins to move upward, and that when the wheel has completed its "jump" and returned to the ground the body has traveled only 40 per cent of its upward path.

In Fig. 493 the lower curve denotes the path which the center

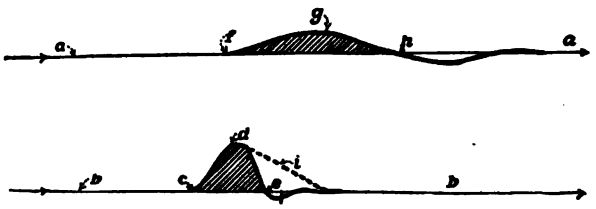


FIG. 493.—Path described by wheel and body.

of the wheel describes when surmounting the obstruction, while the upper curve shows the path pursued by a point of the frame or body just above the center of the wheel. Tests have also shown that when a stiffer spring is used the oscillations of axle and body cease sooner, but that the acceleration of the upward motion of the body is greater, which means more discomfort.

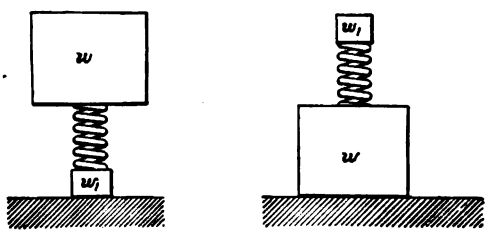


FIG. 494.—Illustrating sprung and unsprung weight.

Relation of Sprung to Unsprung Weight.—Figure 494 illustrates graphically the importance of keeping the proportion of sprung to unsprung weight very great. A comparatively heavy weight is represented by w, while w_1 is a small weight. In the first instances a small weight at the bottom, corresponding here to the axle and the wheel, will have relatively little effect upon the greater weight above, since the latter is so much heavier; the greater weight will be influenced relatively little by what the lighter weight may do. In the second case we have a heavy weight at the bottom, which, if it begins to move upward, will continue to do so, irrespective of the small weight above, and no

matter how weak or strong the spring above may be, it will carry the small weight along with it. Therefore, as far as riding quality is concerned, a lighter unsprung weight will be superior, since it has the least effect upon the sprung weight.

In considering the action of a wheel of a moving car when striking an obstruction over which it rises, so many modifying factors are involved that it is doubtful whether, with the data available at present, a correct and complete mathematical analysis is possible. Still, certain assumptions can be made that more or less approximate true conditions, and a result secured such as will give some idea of the magnitude of the forces involved. When the wheel, moving along a horizontal surface, strikes an obstruction, there is a tendency to reduce the speed of the car because there is imparted to the wheel of the car potential energy that has been acquired at the expense of the kinetic

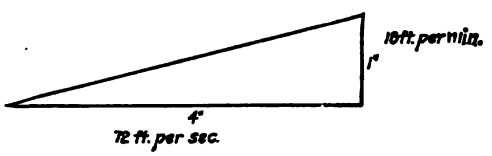


FIG. 495.—Wheel acceleration.

energy of the car. Since the mass of the car is large compared with the unsprung weight per wheel, we can assume as a first approximation, that the car speed remains unchanged while passing over the obstruction. The simplest assumption possible is that the path of the hub of the wheel is a straight line inclined at a definite angle to the horizontal. This is equivalent to assuming that the wheel is traveling up an inclined plane.

When the wheel strikes an inclined plane there is a tendency to retard its motion, since it cannot acquire an upward velocity in zero time. This brings about a changed deformation of the tire and a slight distortion in the supports dragging the axle and the wheel for an instant has a velocity which is slightly less than that of the car. As the distortion in the dragging supports disappears, due to the resiliency of the material, the wheel again catches up in speed and finally when the distortion has completely disappeared the speed of the car and of the wheel is identical and the latter has attained its maximum vertical velocity. From the foregoing it follows that the vertical acceleration is variable.

Let us further assume that the wheel in passing over an obstruc-

tion rises through a vertical distance of 1 in., and that in doing so it travels a horizontal distance of 4 in. (see Fig. 495) and that at this instant the wheel has attained its full vertical speed. The wheel must then have a vertical velocity of 18 ft. per second. Since this vertical velocity V was acquired while the wheel travelled a horizontal distance of 4 in. or $\frac{1}{3}$ ft., the time interval t is $\frac{\frac{1}{72}}{\frac{1}{3}} = \frac{1}{216}$ sec. Then the average acceleration is:

$$a = \frac{v}{t} = \frac{18}{\frac{1}{216}} = 3888 \text{ ft. per second per second}$$

Because the acceleration varies from zero to zero, the actual acceleration for some points of the path must be considerably greater than the value found for the average. Ordinarily the axle will not rise 1 in. in traveling 4 in. horizontally, but this rather unusual case was purposely selected to indicate roughly the magnitude of accelerations that do occur. The upward acceleration of the axle amounting to anywhere near the above figure, will exert a tremendous force against the body and tend to give it a definite upward acceleration and velocity. This upward force of the axle can be reduced by decreasing the unsprung weight and also by some means that permit the axle to move slightly backward with respect to the frame, when subjected to an upward thrust. This is often accomplished in practice by raising the front end of the spring, for instance.

One function of the spring is to decrease the upward velocity and acceleration of the sprung weight, or car body, and to increase the distance of its horizontal travel while in a raised position. The spring thus transforms the vertical distance of the body deflection, either up or down, into a horizontal distance, whenever the wheel surmounts an obstacle or sinks into a rut. In Fig. 493, if the wheel hits the obstruction at *c* it will quickly rise to *d*, that is to say, in a very short horizontal distance, and will soon thereafter drop back to its normal path *bb*. The body above will start to rise at *f*, reach its highest peak at *g* and then slowly descend to the normal path, *aa*. It is evident that the later the curve of the body-path begins to rise, after the wheel starts its upward acceleration, the lower the peak *g*, or the longer the time required for it to reach its maximum height. The greater the horizontal distance *fh* the better will be the riding quality of the car. If no kinetic energy is absorbed by the spring

the area of the shaded section under the lower curve will be approximately equal to that under the upper curve, but the greater the amplitude of the spring deflection and the lower its periodicity, the lower and longer will be the upper curve, described by the body, and the higher and longer the lower curve described by the wheel center. A flexible spring, one with a large amplitude, will permit the axle to fly up very rapidly and to a great vertical height; while a low periodicity will return it to the normal path more slowly, as shown by dotted line *di* (Fig. 493). Even if the upper curve were higher, if it is proportionately longer before returning to normal, a more favorable riding quality will result.

While the formula for finding the periodicity of a spring, or number of vibrations per minute, can be found in any textbook on the subject, let us analyze it by simple practical experiments. If we take four leaves of spring steel, all of which are identical as to length and thickness, and if we hold one end fixed and leave the other free to vibrate, we find that without any load the periodicity will be the same whether we have one leaf or two, three or four leaves together. If the leaves are straight and smooth, so that there is no friction between them, we will also find that the amplitude of deflection is practically the same for the *same load per spring*, as is also the number of vibrations, before the spring finally stops. If we now place a weight of 1 lb. on the free end (see Fig. 496), we find the amplitude of deflection larger but its periodicity lower. For instance, in one case, without any load on the free end, a certain single leaf thin spring, 30 in. long, had a periodicity of 126 vibrations per minute regardless of the number of leaves. When 1 lb. was placed on the free end, the vibrations were reduced to 100; with 2 lb. to 86; with 3 lb. to 78; with 4 lb. to 73, etc. When 1 lb. was placed on the spring composed of two leaves, the periodicity was 112, but with 2 lb. it was 100; with three leaves, 1 lb. reduced the vibrations to only 116; 2 lb. to 108; 3 lb. to 100, etc. This is shown graphically in Fig. 496. With 1 lb. on one leaf, 2 lb. on two leaves, etc., the periodicity as well as the amplitude remains the same.[1] By shortening the leaf to 24 in. the vibrations were increased to 196, without load, while with a 1-lb. load they dropped to 142; with 2 lb. to 120, and with 3 lb. to 110. Two leaves shortened to 24 in. gave

[1] It might be mentioned that these tests were not carried out by the author with extreme accuracy; also the load was not pound weights, but blocks of uniform size and small weight.

a periodicity of 172 with a 1-lb. load; 142 with 2-lb., etc. Three leaves gave a periodicity of 176 with a 1-lb. load; 142 with 3-lb., etc. We can state, therefore that, everything else being equal, a heavier load, which is always accompanied by a larger static deflection, causes a greater fiber stress in the spring; and the heavier the load, or the greater the deflection or the greater the fiber stress, the slower the periodicity.

Let us now consider the effect of friction between the leaves. In the simple experiments covered by *d*, (Fig. 496), rubber bands

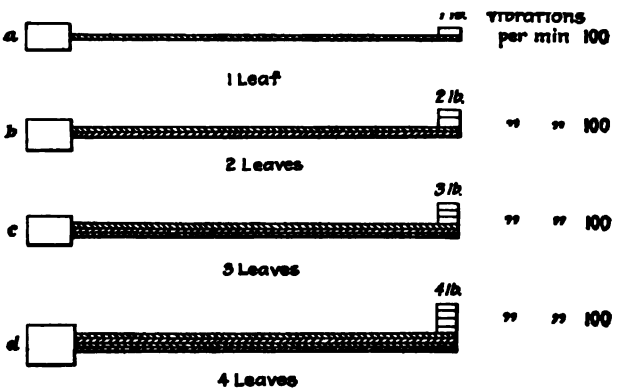


FIG. 496.—Periodicity.

were tied around the springs to create friction between the leaves. We still obtain vibrations at the rate of practically 100 per minute but instead of being able to count the vibrations for a whole minute, they are damped out in about 10 sec. and the static deflection is less than it was without the rubber bands. It is thus evident that friction between the leaves reduces the amplitude of deflection and the number of spring inertia vibrations before the spring comes to rest.

Fundamentals of Periodicity.—We may briefly analyze the fundamental equation governing the time t in seconds, required for one period or one oscillation down and one up. Assume a helical spring suspended vertically and supporting a mass m whose weight is mg lb.; then the mass will come to a position of equilibrium when the restoring force of the spring is equal to mg. According to Hooke's law, if f is the force in pounds required to elongate the spring one unit, then fs will be the force required to elongate the spring s units. Since the spring obeys Hooke's

law, the restoring force tending to bring the mass to its position of equilibrium is proportional to the displacement from the position of equilibrium, and, therefore, if free to vibrate will execute a simple harmonic motion.

The acceleration for a simple harmonic motion is $\frac{4\pi^2}{t^2} s$.

Therefore, the force for the displacement s is $m\frac{4\pi^2}{t^2} s$, but this must be equal to the restoring force. Therefore,

$$m\frac{4\pi^2}{t^2}s = fs, \text{ from which } t^2 = \frac{m4\pi^2}{f} \text{ and } t = 2\pi\sqrt{\frac{m}{f}}$$

The mass and the restoring force govern the periodicity. The restoring force in a leaf spring depends on the length, thickness and width of the leaf and on the modulus of elasticity.

In comparing leaf and coil springs proper allowance must be made for the spring mass involved, but in general the formula governing all vibrating bodies is:

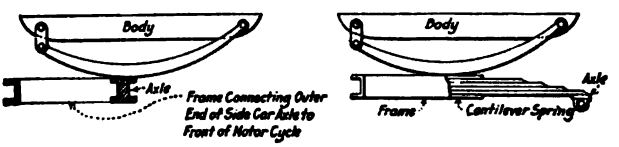
$$t = 2\pi\sqrt{\frac{\text{resistance}}{\text{restoring force}}}$$

It has been found that by attaching a coil spring at the end of a leaf spring, the coil spring does reduce the acceleration of the free end of the leaf spring. When several leaves are used and considerable frictional surface between the leaves is thus introduced, the coil spring will absorb slight vibrations without transmitting them perceptibly to the leaf spring.

This is precisely what happens in practice by the application of coil springs to the ends of leaf springs. They absorb the small rapid vibrations, while the leaf spring is practically at rest, but in the case of large vibrations, the leaf spring deflection is the governing factor. Nevertheless, the deflection of the coil spring is added to that of the leaf spring, and reduces the acceleration of of the "free" end of the leaf spring or that end to which the coil spring is attached.

By substituting a separate leaf spring between the main spring and the axle, better results are obtained. The author's experience in this connection may be of interest. While engaged in redesigning a rather heavy motorcycle with a side-car which among other defects, rode most uncomfortably, he found that the side-car frame was rigidly attached to the axle (see Fig. 497). There were leaf springs 38 in. long between the frame and the

side-car body, but the proportion of unsprung to sprung weight was comparatively large, because the frame rested directly on the axle. There was available also an experimental side-car frame terminating in a short, stiff quarter-cantilever spring attached to the axle. The body was mounted directly on the frame. In testing this on the road it was found to be as bad as an empty truck as far as riding quality is concerned. The body was consequently removed and the first side-car body with semi-elliptic springs placed on the frame, which terminated in the



FIGS. 497 and 498.—Special spring suspension.

cantilever spring, as shown in Fig. 498. This gave a combination of short flat springs between the axle and the frame with the ordinary semi-elliptic springs underneath the body. The first test of this combination was a revelation. It was the unanimous opinion of those who tried it that it rode with greater comfort than any high-priced automobile. This shows the advantage of having an extra spring below the main one. In the first place, this gives an added deflection (although in this case the deflection per unit weight was very small indeed compared with the main spring) and this reduced the periodicity of the body, and secondly the acceleration and velocity of the free ends of the main spring were therefore less.

Thus, to obtain increased comfort, one suggested method is to provide an auxiliary spring, preferably of the leaf type. Another great advantage of the leaf spring is that the inter-leaf friction reduces the vibrations to a very small number before the body comes to rest, while with coil springs the vibrations continue for a long time, and this is to be avoided in car suspension. On the other hand, excessive friction between the leaves interferes considerably with spring deflection and renders the spring much stiffer, thus imparting greater shocks to the sprung weight. It seems that comparatively little friction between the leaves, as for instance when they are lubricated, is sufficient to reduce the total number of vibrations before coming to rest to that required for comfort, except when going at high speed over bad roads,

in which case added friction is advantageous. Friction in the spring shackles also influences the deflection; likewise the angle of the shackle with respect to the spring. If this angle is such that the stress applied to the spring is at right angles to the leaves, it will impart purely flexure stresses to them. If at a greater angle, it will cause, in addition, buckling stresses; and if the angle is smaller, tensile stresses.

Another point worthy of notice is that, with leaf springs composed of a number of leaves, a certain load is required before the spring begins to deflect, due to the inter-leaf friction. This is a serious drawback in both automobile and truck applications; in

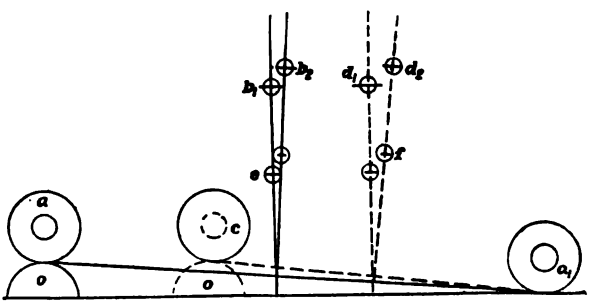


FIG. 499.—Length of wheelbase and center of gravity as affecting passenger comfort.

the first case it renders a touring car less comfortable unless fully loaded, and in a truck it means excessive jars and shocks when empty or only partially loaded. The latter, under such conditions, might as well have no springs, in many cases, since at such loads there is very little, if any, spring deflection.

Center of Gravity and Wheelbase.—The effect of the height of the center of gravity and of the length of wheelbase is illustrated in Fig. 499. Consider first a and a_1, which represent respectively the front and rear wheel of a car having a long wheelbase. Let b_1 represent a point in the body of the car. When the front wheel surmounts an obstruction o, point b_1 will be moved to b_2, due to the angular rise of the front wheel with respect to the normal position of the rear wheel. In this case the backward distance is very small compared with that shown in d_1 and d_2, which represents an analogous case with a short wheelbase car. It is the backward or transverse motion that causes the passenger discomfort, rather than the vertical motion, everything else being equal, and naturally the greater the horizontal distance through

which the passenger is moved, the greater the discomfort. These remarks apply also to the width of tread gauge as affecting side-to-side motion. The illustration moreover discloses what would happen if the points b_1 and d_1 were lowered, or the center of gravity of a car were lowered; in such event the horizontal or transverse motion is reduced when the car wheels rise over an obstruction or sink into a rut.

CHAPTER XV

GEARS AND BEARING LOADS[1]

Spur Gears.[2]—Ordinary spur gears have teeth running parallel to their axis. Helical spur gears, spiral gears or cross gears, have the teeth running at an angle with respect to the axis. A helical tooth of 30° for instance, means that the run of the tooth forms that angle with the axis.

The *pitch circle* of a gear is the circle described by the center of

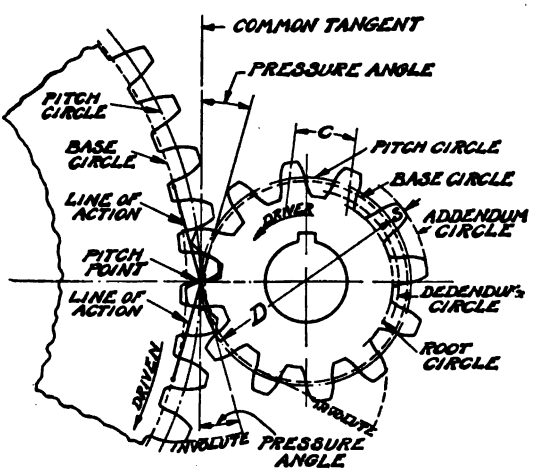


FIG. 500.—Gear wheels.

the line of contact of the teeth as seen in Fig. 500 and the diameter of the pitch circle is called the *pitch diameter*. The circumference of the pitch circle = pitch diameter $\times \pi$. The *pitch* of a gear wheel is a measure of the size of its teeth; this measure may be determined on the pitch circle. The distance from the center of one tooth to the center of the next tooth, measured *on the pitch circle* (not straight across) in inches, is called the *circular pitch C*.

[1] See also pages 128 to 139.

[2] See "The Involute Gear," by the Fellows Gear Shaper Co., and "Formulas in Gearing" and "Practical Treatise on Gearing," by Brown & Sharpe Mfg. Co.

When the number of the teeth in a gear wheel is given for each inch in diameter (the pitch diameter) it is called *the diametral pitch*, that is to say, the number of teeth in a gear wheel divided by the pitch diameter is the *diametral pitch*, or shortly called *pitch*, and this is the measure ordinarily given on all drawings and specifications. Hence when reference is made to pitch, it means $\frac{\textit{number of teeth}}{\textit{pitch diameter}} = \textit{pitch}$ by which is meant *diametral pitch*.

For instance if there are 30 teeth in a gear wheel, and the pitch diameter is 6 in., then the pitch of the gear is $\frac{30}{6} = 5$. If we know the pitch, and the number of teeth, we can easily find the pitch diameter from the above formula. If

P = the pitch (diametral pitch)
N = the number of teeth, and
D = the pitch diameter,

we have

$$P = \frac{N}{D}, \text{ hence } N = PD, \text{ and } D = \frac{N}{P}.$$

If we wish to find the *circular pitch* C when the diametral pitch P is known, we first determine N or D, when it can easily be found the aid of π (which is the ratio of a circle to its diameter and equals 3.14159). Hence

$$C = \frac{\textit{circumference (at pitch circle)}}{\textit{number of teeth}} = \frac{D\pi}{N}, \text{ from which } CN = D\pi,$$

and $\frac{N}{D} = \frac{\pi}{C}$. Since we have $\frac{N}{D} = P$, $P = \frac{\pi}{C}$, and $C = \frac{\pi}{P}$.

For example, if, as before, the pitch is 5, and it is required to find the circular pitch, then $5 = \frac{\pi}{C}$, $C = \frac{\pi}{5} = \frac{3.14159}{5} = .6283$, that is, the circular distance from the center of one tooth to the other, *on the pitch circle*. Figure 500 gives the terms used in connection with gears. It should be noted that the *base circle* is the circle on which the centers of the tooth outlines, or the arcs of the teeth, are formed. The outside circle of a gear wheel, *i.e.*, at the ends of the teeth, is called the *addendum circle*, and the addendum s is the distance of the tooth from the pitch circle to the outside circumference of the gear wheel. For standard involute teeth the length of the tooth above the pitch circle, the addendum s, is made $\frac{1}{\textit{diametral pitch}}$, and this also equals $\frac{\textit{circular pitch}}{\pi}$, thus $s = \frac{1}{P} = \frac{C}{\pi}$. Therefore

the *outside diameter* D' of a gear wheel with standard teeth, (and this the designer must often consider) is equal to the pitch diameter $+ 2s = D + 2s$, and since $D = \frac{N}{P}$, and $2s = \frac{2}{P}$, therefore $D' = \frac{N}{P} + \frac{2}{P} = \frac{N+2}{P}$ (see Fig. 501.) Hence if the number of teeth and the pitch are given, the outside diameter D' can be found very easily. The *dedendum* of a tooth is the working depth below the pitch circle (as far as the tooth from the other gear reaches) and this is equal to the addendum, *i.e.*, the height of tooth above the

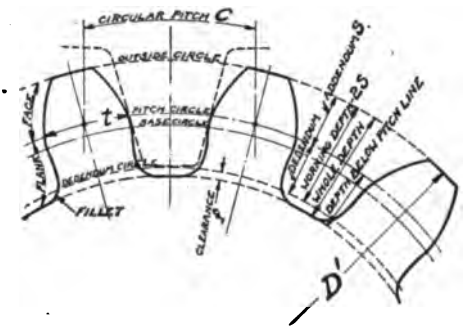


FIG. 501.—Gearing.

pitch circle; thus the dedendum $= s$ (the same as the addendum) and a circle drawn from the center of the gear wheel on the dedendum line or working depth of the tooth is called the *working depth circle* or the *dedendum circle*. Hence the working depth of a tooth is $2s$. Below the working depth of the tooth it is necessary to have some clearance, and a circle drawn at the root of the teeth, *i.e.*, at the bottom of the whole depth of the teeth, is called the root circle. Ordinarily the *clearance* f is made one-tenth the thickness t of the tooth. The *tooth thickness* t on the pitch circle, is one-half the circular pitch, thus $t = \frac{C}{2}$, and in terms of the diametral pitch $t = \frac{\pi D}{2N} = \frac{\pi}{2P}$; (since $\frac{N}{D} = P$, and $\frac{D}{N} = \frac{1}{P}$).

However, when measuring a tooth or laying out teeth, in order to accurately measure the tooth thickness, it is necessary to measure it not on the pitch circle, but straight across, that is to say measure the *chordal thickness* t' of the tooth. This is easily found by trigonometry (see Fig. 502).

If R is the pitch radius ($\frac{1}{2}$ the pitch diameter or $\frac{D}{2}$) then evidently $\frac{1}{2}t' = R$ sine A, and $t' = D$ sine A, (since $2R = D$). The *angle* A is $\frac{1}{4}$ of the circular pitch or

$$\frac{1}{4} \text{ of } \frac{360°}{\textit{number of teeth}} = \frac{90°}{N}.$$

The *face* of the tooth as seen in Fig. 501, is the part of the tooth outline extending above the pitch circle.

The *flank* of a tooth is the portion of the tooth extending from the pitch circle to the root.

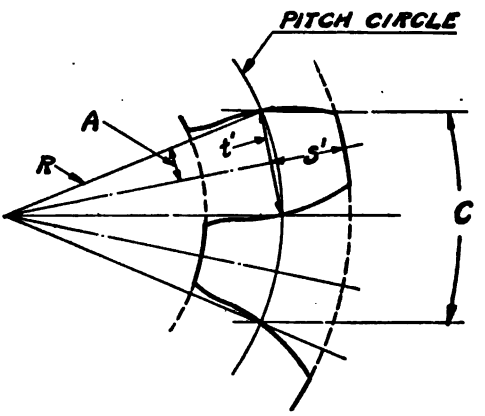


FIG. 502.—Chordal thickness of tooth.

The *fillet* of a tooth is the rounded corner where the flank runs into the root circle.

It was mentioned before that the base circle was the line on which the center of the outline of the tooth arc was located. By drawing a line from the base line of one gear wheel to that of the other, we have the line of action on which contact between the teeth takes place. In the standard involute tooth, as originated by the Brown & Sharpe Mfg. Co., the base lines are so drawn that the *angle of action* or the *angle of pressure* (since the pressure from one tooth will be transferred to the other on this line) is $14\frac{1}{2}°$ with regard to the common tangent of the two gears, (see Fig. 500). In the involute form advocated by the Fellows Gear Shaper Co., the angle of action or angle of pressure is 20° and this is largely used in automobile practice today. Also the addendem is made shorter, hence the tooth is called the *stub-tooth*. In the stub tooth system the pitches are designated by two figures

and are ordinarily as follows; 4/5, 5/7, 6/8, 7/9, 8/10, 9/11, 10/12, 12/14. Frequently the numbers are written one after the other, for instance pitch 5–7. The first number designates the diametral pitch as in the standard tooth system but the second number indicates the height of the tooth. For instance, if the pitch is 5–7, the diametral pitch is 5, and the addendum is 1/7 in. in height. As an example, say a spur gear wheel has 30 teeth, and 5/7 pitch. The pitch diameter $D = \frac{N}{5} = \frac{30}{5} = 6$. The outside diameter of this gear wheel $D' = D + 2 \times 1/7 = 6 + 2/7 = 6.285$ in. An advantage claimed for the *stub-tooth* over the standard 14½° tooth is that by increasing the pressure angle and shortening the tooth, the sliding action between the teeth is reduced as compared to that of the standard 14½° tooth and there is proportionally a greater rolling action and hence less friction between the teeth. Furthermore, by shortening the tooth and increasing the pressure angle it is possible to widen the tooth at the base and therefore greater strength is obtained with the stub tooth for the same pitch. Hence with the stub-tooth it is claimed that a finer pitch is possible for a given strength, and the greater strength is especially present when the number of teeth in the gear wheel is small, as usually required in motor vehicle engineering.

In motor vehicle engineering it is often necessary to use other forms of teeth in spur gearing besides straight teeth, as for instance, in the timing gears for engines, where gears with helical or twisted teeth are frequently employed, *i.e.*, the teeth, instead of running parallel with their axes, run at an angle (in timing gears usually about 30°). The advantage of helical or twisted gears is that there are more teeth than one constantly in mesh and therefore it is possible to obtain more silent and smoother running than with straight teeth; with the latter, the entire length of tooth comes into engagement at one time, while with the helical gear it comes into engagement gradually. When the axes of a pair of twisted or spiral gears run parallel, the teeth of the gears must have opposite hand spirals. When their axes are at right angles, both have the same hand spiral teeth. *The lead* or *spiral lead* is the distance which the spiral advances in one revolution. The lead of any spiral $= \frac{\textit{circumference (pitch circle)}}{\textit{tan of angle } Y.}$ $= \pi \times$ pitch diameter $\times$ cotangent of angle. The tan-

gent of the angle of the spiral $Y = \frac{circumference\ (pitch\ circle)}{lead}$

With helical or twisted gears, the pitch may be expressed in three different ways, *viz.*, normal, circular and axial pitches. (See Fig. 503.)

The normal pitch n is the shortest dimension from the center of one tooth to the other. The circular pitch C is measured on a line parallel to the sides of the gear wheel, while the axial pitch x is the distance measured parallel to the axis.

When the axes of two gear wheels are parallel and the teeth helical, they are usually called twisted gears; when their axes or shafts are not parallel, gears with twisted teeth are usually called spiral gears.

Bevel Gears.—Almost every final drive used on passenger cars

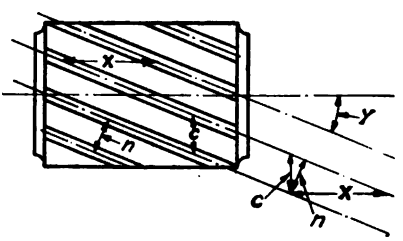


FIG. 503.—Pitches of helical gears.

and some forms used on trucks, embody bevel gears. In fact it is only with the "worm drive" and "friction drive" that bevel gears can be dispensed with. They are used when shafts meet at an angle. In automobile rear axles the angle is 90°, that is to say, the propeller shaft makes an angle of 90° with the rear axle. Bevel gears must be designed in pairs in order that their teeth may properly mesh, for even the same pitch bevel gears will not mesh except with a mate of a given number of teeth. In bevel gears (as well as in spur gears) the smaller gear is called the "*pinion*," and the larger, the "*gear*," "*gear wheel*" or "*ring gear*." The designing of bevel gears is in many respects the same as that of spur gears. In the design of spur gears the pitch circle forms an imaginary cylinder and is the line on which the two gears roll, so to say. In bevel gears we imagine a pitch cone circle to form the line of rolling contact between the two gears. Figure 504 shows the bevel gear pitch cones.

In bevel gears the *pitch*, *pitch circle* and *pitch diameter*, are always figured at the larger end of the cone or bevel, but they are found in the same manner as for spur gears. (See Fig. 505.)

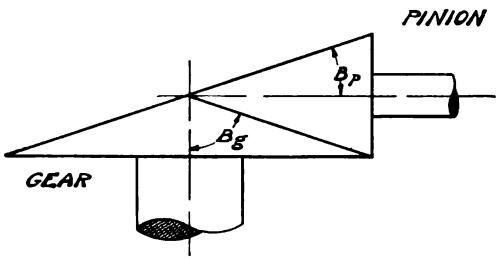


FIG. 504.—Bevel gear pitch cones.

In the Fig. D and D_1 are the pitch diameter of the gear and pinion respectively, and are formed in the same manner as for spur gears. The diameter or the distance across the pitch circle

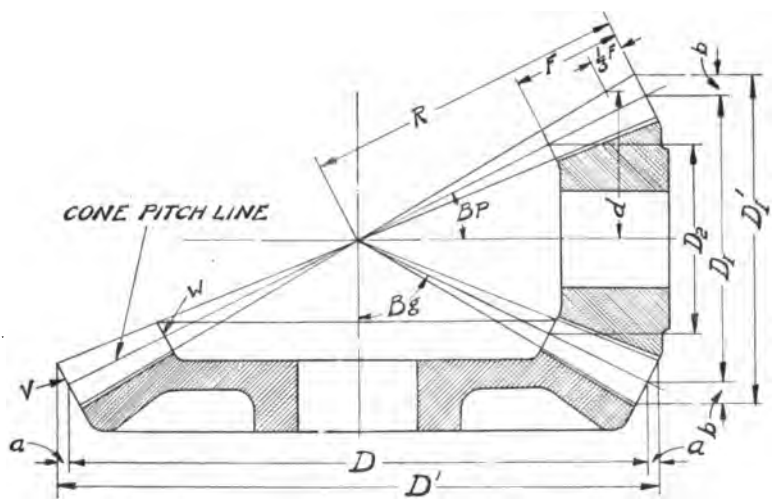


FIG. 505.—Bevel gears.

on the smaller end of the cone, of the pinion or the gear, as the case may be, is called the inner or smaller pitch diameter; it is marked D_2 in the drawing for the pinion. B_g and B_p are the *pitch angle* (sometimes called the center angle or angle of edge) of the gear and the pinion respectively. As may be noted, the cone pitch line, as well as the addendum line and clearance line

of the teeth of both pinion and gear, run to a central point. Therefore, when the addendum, dedendum and clearance are determined on the outside of the cone, by drawing lines to the center, we can measure the height, etc., of the tooth on the inner end of the cone. The sides v and w at the ends of the face width, are at right angles to the cone pitch line. The outside diameter D' and D'_1 are obtained in a different manner from those for spur gears on account of the angle formed, but these dimensions the automobile engineer must be able to determine on account of the necessary housing clearance as for instance in the rear axle housing. $D' = D + 2a$, and $D'_1 = D_1 + 2b$; and $2a = 2s \cos B_g$, and $2b = 2s \cos B_p$, s being the addendum. To find the pitch angles we have the formula tang $B_p = \frac{N_p}{N_g}$, and tang $B_g = \frac{N_g}{N_p}$. N_p being the number of teeth of the pinion and N_g that of the gear. Sometimes it is also necessary to find the pitch cone radius R (often called the apex distance from the pitch cone circle) or the outside cone radius. $R = \frac{D}{2 \sin B_g} = \frac{N_g}{2P \sin B_g}$. If we consider the pinion we take pitch diameter D_1, number of teeth N_p, and angle B_p of the pinion, when the result R will be found to be the same for the pinion as for the gear. These dimensions are for bevel gears whose shafts form an angle of 90°, and for standard involute teeth. If the tooth is made shorter than the standard tooth, some of the dimensions will slightly vary. When the tangent has been found we can determine the angle from any book giving a table of trigonometrical functions, and knowing the angle, we can find the sine from such tables. By drawing the gears on a large scale the designer can usually measure direct, the dimensions necessary for his purposes. The face width F is not made greater (usually less) than $\frac{R}{3}$ or $\frac{5C}{2}$, whichever gives the lesser value (C being the circular pitch). The mean pitch radius d, which is used for determining the bearing pressure, is not the distance from the axis (the center of the shaft) to the center of the tooth, on the cone pitch line, but it is usually taken nearer to the outside of the tooth (nearer to the pitch diameter) because the load on the tooth varies with the radius. The mean load point on the tooth is therefore located

at a greater radius. If F is the face width, and R the cone pitch radius, the mean pitch radius d for the purpose of bearing calculations may be located at a point, as shown in the drawing, R-⅓ F. (For complete designs of bevel gears the author would refer readers to books on this subject.)

Strength of Gear Teeth.—The strength of gear teeth is usually determined by the formulas proposed by Mr. Wilfred Lewis, which can be found in any engineering handbook or treatise on gears. In automobile practice, the variable load and *shock* necessitates the employment of stronger gears than in most other type of machinery, for this reason we have given a number of detail drawings, and indicated the pitch and width of the gears of many standard cars and trucks, which actual practice has demonstrated to be satisfactory.

In the spiral bevel gear[1] the tooth is curved, instead of straight, and the curve, if continued, would intersect the cone center of the gears. This distinguishes the spiral bevel from the skew bevel gear in which the teeth run straight, though at an angle, and do not intersect the cone center. The greatest advantage of the spiral type of bevel gears is their silent running, even when not cut so accurately, and also, during their entire life, it is easier to keep them quiet, as the adjustment need not be so accurate as with straight bevel gears. The chief reason for their quiet operation is due to the overlapping of the teeth, more than one tooth always being in contact and the whole tooth does not come into, or out of, contact at once.

The action of the spiral type of bevel is mostly straight rolling as is the case with the straight tooth spur gear. Another marked advantage of the spiral bevel is the range of endwise adjustment of the pinion without materially reducing the amount of tooth bearing surface and without introducing noise. A number of tests which have been made on a testing machine to determine the amount of endwise motion with different combinations and pitches, showed that the pinion may be moved forward and backward .015 in. (.007 to .008 each way from the flush position) and even beyond this adjustment the spiral bevel gear remained quiet and only gradually became noisy thereafter. Similar tests carried out with straight tooth bevels showed a range of endwise adjustment of only one-half the amount of that for bevel gears and beyond this distance they immediately become very

[1] From A. L. STEWART, *S.A.E. Transactions*, 1915, Part II.

noisy. Hence while a straight tooth bevel may be quiet if perfectly cut and if in perfect alignment under all conditions, it is difficult to keep it so in practice.

Another advantage of the spiral type bevel gears is the higher permissible ratio between pinion and gear, and hence the gear may be made smaller; this decreases the diameter of the big gear, which means a smaller housing and more ground clearance. The spiral bevel is stronger than the straight tooth bevel since more teeth are in contact at one time, hence a finer pitch may be employed. On the other hand, for the same power transmitted, other things being equal, the load on the spiral teeth is greater than on straight teeth on account of the additional thrust load imposed by the spiral. With a spiral of 30°, the total load on the tooth is about 15 per cent greater than the corresponding tooth load for straight tooth gears, but the fact that more teeth than one are in contact, more than compensates for this 15 per cent increase, and there is in fact a smaller load and thus less strain on each tooth.

FIG. 506.—Spiral bevel gears (right hand). (*Cut by the Gleason Works.*)

It has been found that the efficiency of these gears is as high as that of straight tooth bevels, even though there is an added end thrust on the pinion bearings. In automobile work, ball bearings are generally used to take up this end thrust and the friction loss in such bearings is very small.

Direction of Spiral and Thrust.—The spiral may be right hand or left hand in direction. In speaking of the direction of the spiral, this is named from the direction of that of the pinion; it should be remembered that the spiral of the gear wheel is always the opposite to that of the pinion, *i.e.*, if the pinion is cut with right hand spiral, the mating gear wheel will be cut with a left hand spiral, yet the pair of gears in mesh will be called "right hand spiral."

Figure 506 shows a set of right hand spiral bevel gears, cut by the Gleason works of Rochester; it derives its designation, right hand, from the fact that the spiral of the pinion runs like the thread of a screw. When turning it to the right, the front end (small end) of the pinion will roll into contact first. There is no difference as far as efficiency or quiet operation is concerned, whether the spiral is right or left handed, but it does make a difference to the direction of the thrust load, *i.e.*, end thrust of the pinion. The thrust due to the spiral bevel gear is much larger than that of the straight tooth bevel and it varies in amount with the angle of the spiral. It also changes in direction with the direction of rotation of the pinion, for with a right hand spiral there will be a tendency for the pinion to pull into the gear wheel on forward drive, and a thrust back, away from the gear, on the reverse.

With a left hand spiral on the forward drive the thrust is backward, away from the gear, on the reverse speed, it is "into the gear." Combining the usual thrust of the straight bevel tooth with that of the thrust caused by the spiral, the following are the average results for ratios between 3 : 1 and 7 : 1:—

Angle of spiral.....	20°	25°	30°	35°
Right hand, forward.....	−.29	−.44	−.56	−.69
Right hand, reverse.....	.41	.59	.72	.85
Left hand, forward.....	.41	.59	.72	.85
Left hand, reverse.....	−.29	−.44	−.56	−.69

The results given in decimals, represent the thrust as a percentage of the transmitted tooth load. The positive values indicate the thrust away from the gear, while the negative values indicate pull into the gear. From the table it will be seen that for a 30° spiral angle, for instance, and a right hand spiral, the thrust for forward drive is 56 per cent of the transmitted tooth load and tends to screw the pinion into the gear; for reverse drive it is 72 per cent and thrusts the pinion back from the gear. The designer has the choice between the right and left hand spiral, which is based on the general conditions of the design. If the bearings are so arranged as to be held firmly against end movement, the right hand spiral may be used in order to make the thrust load smaller under forward driving conditions. On the other hand, if there is a possibility that the continued inward pull of the pinion into the gear will cause play in the bearings,

or if the type of bearing used is not so well adapted for resisting the thrust in this direction, the left hand spiral is the more satisfactory.

The right hand spiral causes a smaller thrust load under the forward drive than the left hand spiral, and as the car is driven backwards for only very short durations, smaller bearings may be employed. The left hand spiral is chosen at times, when changing from straight tooth bevels to spiral bevels, because the bearings provided with straight bevels for taking the negative thrust or the pull toward the gear, are of smaller capacity than those taking the positive or outward thrust; in such cases the left hand spiral involves less changes in design. Other conditions affecting the choice of direction of the spiral, are lubrication, the question whether the in-pull is liable to loosen the pinion from its taper seat (when they are not made integral) and the possibility of the pinion having enough end-play to wedge into the gear and start cutting. As a rule a spiral angle of 30° is used in automobile work, for 4-, 4½- and 5-diametral pitch and the usual face width of gear. For a finer pitch, or an extra long face, a smaller angle is sometimes satisfactory.

The design of spiral bevel gears is conducted along the same general line as straight tooth bevels.[1] The face length must not exceed ⅓ of the cone distance; in automobile driving gears ¼ of the cone distance has been found to be a good practical limit.

The tooth proportions used on spiral bevels will be different than those used on similar straight tooth gears for the reason that the pressure angle of a spiral bevel gear tooth is greater than that of a corresponding straight tooth bevel gear, where the same cutter pressure angle is used in each case.

The spiral angle should be chosen to give an overlap of the teeth in all cases. The teeth should overlap one another by 1¼ to 1½ times the circular pitch. The working depth of the spiral bevel gear is 1.7 over the diametral pitch. A clearance of not less than .06 times the circular pitch is used. The number of teeth in the spiral bevel gear can be made less than is practicable for straight bevels due to the overlapping nature of the teeth, but in deciding upon the number of teeth to be used in either straight gears or spiral gears, it is well to remember that efficiency and wearing qualities are better with larger numbers of teeth than with small numbers. For spiral bevels, the Gleason

[1] As recommended by the Gleason Works.

works have found 10 teeth to be the minimum practical number for a satisfactory operation in quantity porduction. When extreme quietness is essential, not less than 12 teeth are recommended, though a smaller number of teeth can be used to reduce the size of the installation, but the design, mounting and manufacture of such gears must be given special consideration.

Bearing Pressures in Bevel Gears.—Suppose we take an engine developing 40 hp. at 1,500 r.p.m. The maximum torque T of such an engine may be taken as 1,600-lb. inches. (See Vol. I, "Engines," chapter on Torque.) Assuming the mean pitch radius d of the pinion to be 1.2 in., (for the purpose of bearing calculations d is greater than the mean pitch radius, for the load on the tooth varies with the radius and the mean load on the tooth is therefore located at a greater radius, as was shown before) to find the tangential pressure P, at the tooth, we have the formula (see page 133), $P = \frac{T}{d} = \frac{1{,}600}{1.2} = 1{,}333$ lb. (see Fig. 507).

Assume that the gears have "standard" teeth with a pressure angle of 14½°, and a friction angle of 3°, (see page 129), or a total of 17½°, and call this angle A (cos of 17½° = .9537). The pressure P_1 (*i.e.*, the pressure or the force normal to area of tooth contact) between the gear teeth, as a result of the pressure angle A is, $P_1 = \frac{P}{\cos A} = \frac{P}{\cos 17\frac{1}{2}^\circ} = \frac{1{,}333}{.9537} = 1{,}400$ lb. approximately (to find the direction of this pressure or force P_1 see Figs. 507 and 508).

In straight spur gears this would be the radial pressure or force exerted at the center of the pinion, but since the teeth of the bevel pinion do not run parallel with its axis (with its shaft) this force will be exerted at an angle against the ring gear wheel, and the reaction will therefore act at an angle against the center of the pinion, as seen from Figs. 507 and 508. Since we are interested in finding the radial force against the center of the pinion, at right angles to the pinion shaft, we must first decompose this force into its two components, P and P_2 (since P_1 is composed of P and P_2). P is known already and $P_2 = P \tan A$, (since $\tan A = \frac{P_2}{P}$). This force P_2 exerts a certain pressure at an angle, against the ring gear as shown in Fig. 509. The angle being equal to the pitch angle B (as seen from this figure the angle between P_2 and T_g is the same as the pitch angle B, since P_2 is at

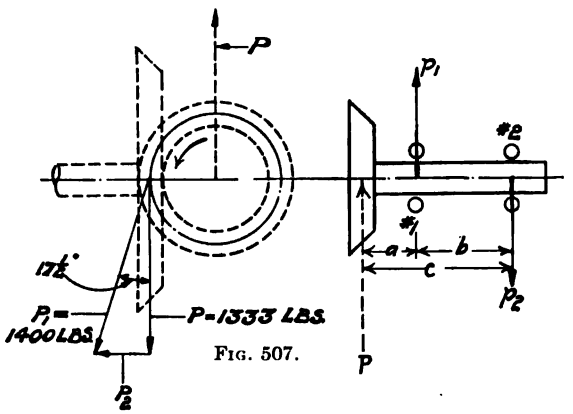


FIG. 507.

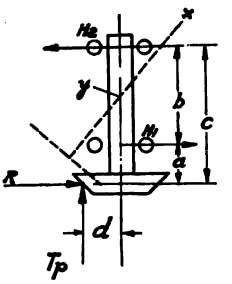


FIG. 510.

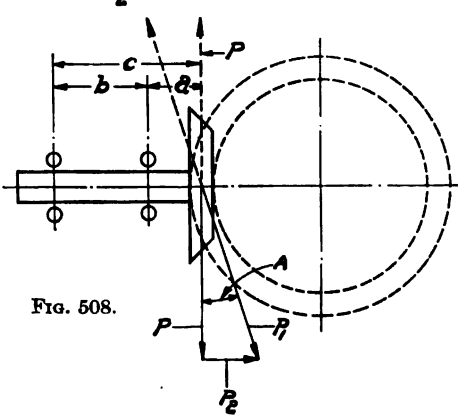


FIG. 508.

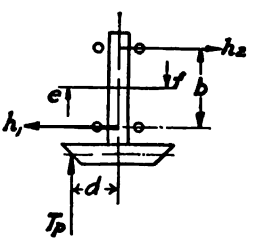


FIG. 511.

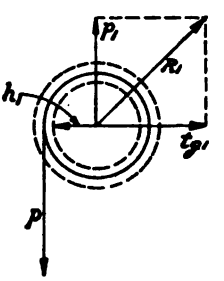


FIG. 512.

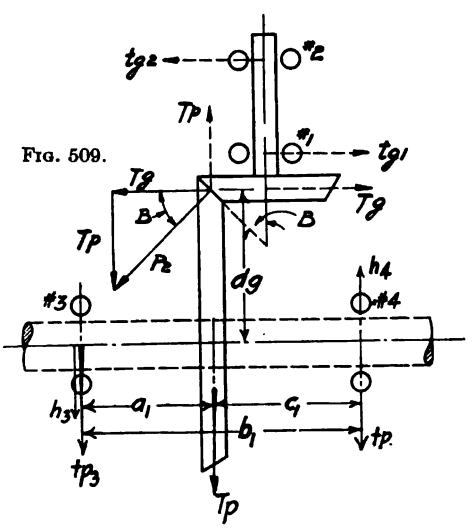


FIG. 509.

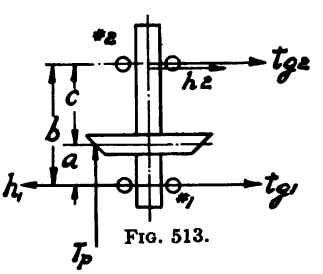


FIG. 513.

FIGS. 507 to 513.—Bearing loads due to bevel gear drives.

right angles to the teeth as shown, and T_g is at right angles to the axis of the pinion), therefore, the horizontal pressure or the force T_g which causes a thrust against the ring gear, in line with its axis is, $T_g = P_2 \cos B$. The reaction of this force T_g (shown dotted on the right) is the radial pressure against the center of the pinion, since action and reaction are equal to each other and opposed in direction. Next we determine the radial pressure T_p against the ring gear, the reaction of which (T_p shown dotted) is the thrust along the axis of the pinion shaft. From Fig. 509, we write $T_p = P_2 \sin B$. The resultant radial load R against the center of the pinion, is therefore the resultant of the force P acting vertically, shown dotted in Fig. 507 and the force T_g (see Fig. 509) acting horizontally, and by the well known formula for right angle triangles, $R = \sqrt{P^2 + T_g^2}$. Suppose the bearings supporting the pinion were located as shown in Fig. 510; the reaction H_1 on bearing *#1*, being at a distance a from the mean pitch circle of the pinion, and reaction H_2 (on bearing *#2*, see Fig. 507), c distant; the distance between the two bearings being b. By taking moments about H_2, we have $Rc = bH_1$ hence $H_1 = \frac{Rc}{b}$ = the resultant radial load on bearing *#1*, and $Ra = bH_2$, therefore, $H_2 = \frac{Ra}{b}$ = the resultant radial load on bearing *#2*. (See pages 16 and 25 for moments of forces.) In addition, when accuracy is required, we must consider the radial load arising from the effect of the thrust load T_p against the pinion (see Fig. 510). This force T_p is exerted against the pinion at a distance d from its center, hence it will tend to make the pinion shaft assume a position as shown by the dotted lines x. That is to say, the force T_p will tend to push the left side of the pinion backward and to the left, and will tend to cause the pinion shaft to swing around on center y midway between the two bearings. The two arrows e and f, (Fig. 511), show the tendency of T_p to swing the top end of the shaft to the right, while at the bottom bearing it will tend to move to the left, as indicated by arrows h_2 and h_1 respectively. By comparing these arrows with t_{g1} and t_{g2}, of Fig. 509, it will be noted that they act in opposition to the radial load on the bearings due to T_g (shown dotted). The total moment tending to tilt the pinion and swing the shaft around to position x, is $T_p \times d$ (Fig. 511); this will impart pressure or force h_1 and h_2 equal in magnitude on

the two bearings, b distance apart (in opposition to that due to T_g). The pressures on the bearings due to T_g are designated t_{g1} and t_{g2}. Taking moments about h_2 (Fig. 511) we have $T_p \times d = h_1 \times b$, hence $h_1 = \frac{T_p \times d}{b}$; similarly by taking moments about h_1, we obtain $h_2 = \frac{T_p \times d}{b}$. These values have to be deducted from the pressures caused by T_g. We will embody this reduction in the original formula and denote the total radial pressure R_1 on bearing *#1* next to the pinion, and R_2 that on bearing *#2*. Remember that: The tangential pressure P is vertical, and T_g horizontal, thus these two forces act at right angles to each other; h_1 and h_2 act horizontally but in an opposite direction; and that the reactions of forces act all in the opposite direction to such forces. In other words if a shaft presses in a downward direction against a bearing, with a force P, the bearing will press against the shaft in an upward direction, but with the same amount of pressure or force P. In Fig. 507, the reaction due to P, p_1 and p_2 at bearing *#1* and *#2*, respectively. Looking at Figs. 507 to 512 to find pressure or reaction p_1 on bearing *#1*, due to the tangential force (vertical pressure) P, we have $P \times c = p_1 \times b$, and $p_1 = \frac{P \times c}{b}$. Due to horizontal force T_g, we have $T_g \times c = t_{g1} \times b$, and $t_{g1} = \frac{T_g \times c}{b}$. The radial force h_1 opposed to t_{g1} due to the thrust T_p is as shown before, $h_1 = \frac{T_p \times d}{b}$; h_1 is opposed to t_{g1} (see Fig. 512) (and must be deducted therefrom) but they both are at right angles to p_1; hence the resultant bearing load $R_1 = \sqrt{p_1^2 + (t_{g1} - h_1)^2}$
In a similar manner we find for bearing #2.
The vertical pressure

$$p_2 = \frac{P \times a}{b},$$

due to T_g, we have

$$t_{g2} = \frac{T_g \times a}{b},$$

and for the radial pressure h_2 (opposed to t_{g2}) due to the thrust T_p, $h_2 = \frac{T_p \times d}{b}$; the resultant radial load or pressure on bearing *#2*, $R_2 = \sqrt{P_2^2 + (t_{g2} - h_2)^2}$.

Attention is drawn to the fact that when the pinion is sup-

ported between two bearings, (instead of in front of two bearings as shown) *i.e.*, if *#1* bearing were located on an extension of the shaft in front of the pinion as seen in Fig. 513. In such event the horizontal pressure or load t_{g2} on bearing *#2* due to the horizontal force T_g would be exerted in a reverse direction. Thus t_{g2} would be on the same side as t_{g1} and h_2 would have to be added to t_{g2}. The radial load on *#2* bearing would then be

$$R_2 = \sqrt{p_2^2 + (t_{g2} + h_2)^2}.$$

The load on *#1* bearing would remain the same. By marking the distance between the bearings b, and the distance between the pinion and the first bearings a as shown, the letters in the formula for finding the bearing loads will be the same as before.

The thrust load against the pinion is T_p, and this is taken up either by bearing *# 1* or bearing *# 2*. The present tendency is to let bearing *# 2* take the entire thrust (formerly the reverse practice was largely in vogue) while *# 1* is used for the radial load only, and this of course is much higher here than in bearing *# 2*. Since the thrust on the bearing acts at right angles to the radial load we must again find the resultant to obtain the total load on this bearing. If *# 2* takes the thrust, then the resultant $R_{t2} = \sqrt{R_2^2 + T_p^2}$.

Load on Ring Gear Bearings.—Since the action and reaction are the same in magnitude and reversed in direction, the radial force against the ring gear will evidently be the resultant of the horizontal thrust load T_p against the pinion (see Fig. 509) and the vertical load P (see Fig. 507) whose direction is at right angles to T_p. Taking moments we have on ring gear bearing *# 3* (Fig. 509) due to P: $P \times c_1 = p_3 \times b_1$; $p_3 = \dfrac{P \times c_1}{b_1}$, ($p_3$ being the radial (vertical) load due to P). Due to T_p, we have: $T_p \times c_1 = t_{p3} \times b_1$, and $t_{p3} = \dfrac{T_p \times c_1}{b_1}$. The radial load h_3, due to the tilting tendency of the thrust T_g against the ring gear is found as follows: $T_g \times d_g = h_3 \times b_1$, hence $h_3 = \dfrac{T_g \times d_g}{b_1}$. d_g being the mean pitch radius of the larger gear which is determined in a similar manner as that for the pinion; (or it may be determined from d as follows: $d_g = d \times$ reduction ratio); h_3 in this case is added to t_{p3} since the left end of the shaft will tend to be tilted in

the same direction as the force T_p, as seen from Fig. 509. Hence the resultant radial load on bearing # 3 will be

$$R_3 = \sqrt{p_3^2 + (t_{p3} + h_3)^2}.$$

The thrust on this bearing acting at right angles to the radial load is T_g. Hence the total resultant load on this bearing $R_{t3} = \sqrt{R_3^2 + T_g^2}$. Bearing # 4 only carries radial

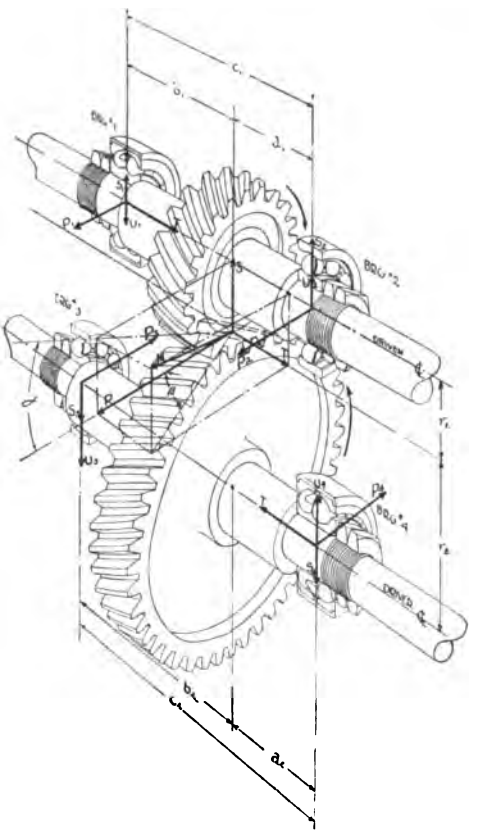


FIG. 514.[1]—Bearing loads of helical spur gears.

load. By taking moments about the left bearing we find radial load p_4 (due to P) as follows: $P \times a_1 = p_4 \times b_1$; hence $p_4 = \frac{P \times a_1}{b_1}$; due to T_p we have $T_p \times a_1 = t_{p4} \times b_1$ and $t_{p4} = \frac{T_p \times a_1}{b_1}$. The radial load h_4 due to the tilting action of the

[1]From The New Departure Mfg. Co.

thrust T_g is: $h_4 = \frac{T_g \times d_g}{b_1}$, and this is deducted from t_{p4}. The resultant radial load is therefore $R_4 = \sqrt{p_4^2 + (t_{p4} - h_4)^2}$. or if h_4 is greater than t_{p4}, $R_4 = \sqrt{p_4^2 + (h_4 - t_{p4})^2}$.

In helical bevel gears there is still another factor which has to be considered, namely the helix angle of the teeth which is usually 30°. This introduces another factor, *i. e.*, another component to the forces before found. But by means of the tables given, on page 445, the bearing pressures may be found quickly and with sufficient accuracy for most practical purposes.

In Chap. VI Transmission Gears, the bearing loads due to straight spur gears were given. Figure 514 gives a diagram, to accompany the following description, to find the bearing loads due to helical spur gears when the shafts are parallel.

Loads Due to Helical Spur Gears (Shafts Parallel).[1]—In this type of gearing, there is, due to the helix angle, an end thrust which must be resisted by the bearings. In Fig. 514, bearing *# 1* is a double row bearing, designed to take radial load and thrust in either direction, and must be mounted with both races fixed, while *# 2*, a single row bearing, designed to take radial load only, will be mounted with the inner race fixed, and outer race free to move endwise. Bearing *# 1*, then, will take care of all the end thrust in the shaft of the small gear.

Similarly, for the large gear shaft, Bearing *# 4* will take the thrust load and its portion of the radial load, while Bearing *# 3*, whose outer race is free to move endwise, will take only its portion of the radial load and no thrust load whatever.

$$\text{Torque} = Q = \frac{HP \times 63{,}025}{n} \text{ in pound-inches}$$

where hp. = hp. transmitted
and n = revolutions per minute of shaft.

P = force tangential to pitch circle of driving gear $= \frac{Q}{r_2}$ lb.

where r_2 = pitch radius of driving gear in inches.

N = force normal to area of tooth contact.

T = component parallel to gear axes = thrust component = $P \tan \beta$, where β = helix angle.

S = separating force = $P \tan a$

where a = tooth pressure angle.

[1] The following discussion and Figs. 514 and 515 are from The New Departure Mfg. Co. publications.

(Above we have assumed a to be measured in a plane perpendicular to shafts; if, as is sometimes done, a is measured in a plane normal to the tooth, then S becomes $\frac{P \tan a}{\cos \beta}$)

Load on Bearing # 1

Due to $P = \frac{P \times a_1}{c_1} = P_1$

Due to $S = \frac{S \times a_1}{c_1} = S_1$

Radial load due to thrust $T = \frac{T \times r_1}{c_1} = U_1$

(This radial load is due to the fact that the thrust is not applied at the axis of the bearing, but at a distance r_1 from it. Thus a moment Tr_1 is set up which must be resisted by the bearings.)

Total radial load $= \sqrt{P_1^2 + (S_1 - U_1)^2} = R_1$

Thrust load $= T$

Total load on Bearing # 1 $= \sqrt{R_1^2 + T^2} = L_1$

Load on Bearing # 2

Due to $P = \frac{P \times b_1}{c_1} = P - P_1 = P_2$

Due to $S = \frac{S \times b_1}{c_1} = S - S_1 = S_2$

Radial load due to thrust $T = \frac{T \times r_1}{c_1} = U_1 = U_2$

Total load on Bearing # 2 $= \sqrt{P_2^2 + (S_2 + U_2)^2} = L_2$

Load on Bearing # 3

Due to $P = \frac{P \times a_2}{c_2} = P_3$

Due to $S = \frac{S \times a_2}{c_2} = S_3$

Radial load due to thrust $T = \frac{T \times r_2}{c_2} = U_3$

where r_2 = pitch radius of large gear.

Total load on Bearing # 2 $= \sqrt{P_3^2 + (S_3 + U_3)^2} = L_3$

Load on Bearing # 4

Due to $P = \frac{P \times b_2}{c_2} = P - P_3 = P_4$

Due to $S = \frac{S \times b_2}{c_2} = S - S_3 = S_4$

Radial load due to thrust $T = \frac{T \times r_2}{c_2} = U_3 = U_4$

Radial load on Bearing # 4 = $\sqrt{P_4^2+(U_4-S_4)^2} = R_4$

Thrust load = T

Total load on Bearing # 4 = $\sqrt{R_4^2 + T^2} = L_4$

Loads Due to Worm Gears.—Referring to diagram of Fig. 515 the thrust on the worm axis is resisted by Bearing # *2*, which is mounted with both inner and outer races clamped. Bearing # *1* is mounted with the outer race free to move endwise, and therefore, takes no thrust.

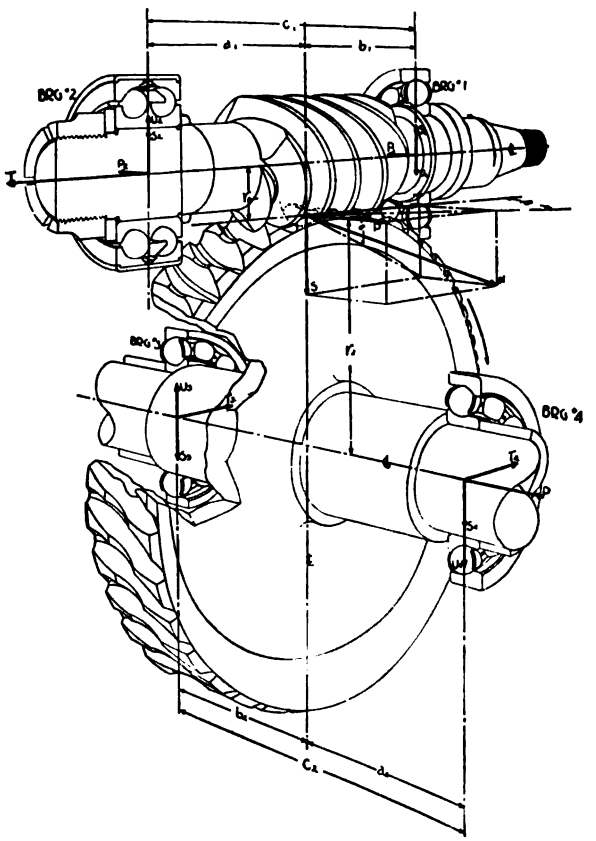


FIG. 515.—Bearing loads of worm gears.

The worm wheel shaft is fixed between two angular contact bearings mounted "opposed." Bearing # 4 takes the thrust load in this shaft.

$$\text{Torque} = Q = \frac{HP \times 63{,}025}{n} \text{ in pound-inches}$$

where HP = hp. transmitted

and n = revolutions per minute of shaft.

P = force tangential to pitch circle of worm $= \frac{Q}{r_1}$

where r_1 = pitch radius of worm in inches.

T = force driving worm wheel = thrust in direction of worm axis $= \frac{P}{\tan \beta}$

where β = lead angle of worm.

S = force tending to separate worm and worm wheel = $\frac{P \tan a}{\tan \beta}$

where a = pressure angle of tooth.

Load on Bearing # 1

Due to $P = \frac{P \times a_1}{c_1} = P_1$

Due to $S = \frac{S \times a_1}{c_1} = S_1$

Radial load due to thrust $T = \frac{T \times r_1}{c_1} = U_1$

Load on Bearing # 1 $= \sqrt{P_1^2 + (S_1 - U_1)^2} = L_1$

Load on Bearing # 2

Due to $P = \frac{P \times b_1}{c_1} = P - P_1 = P_2$

Due to $S = \frac{S \times b_1}{c_1} = S - S_1 = S_2$

Radial load due to thrust $T = \frac{T \times r_1}{c_1} = U_1 = U_2$

Radial load on Bearing # 2 $= \sqrt{P_2^2 + (S_2 + U_2)^2} = R_2$

Thrust $= T$

Total load on Bearing # 2 $= \sqrt{R_2^2 + T^2} = L_2$

Load on Bearing # 3

Due to $S = \frac{S \times a_2}{c_2} = S_3$

Due to $T = \frac{T \times a_2}{c_2} = T_3$

Radial load due to thrust $P = \frac{P \times r_2}{c_2} = U_3$

where r_2 = pitch radius of worm wheel in inches.

Total load on Bearing # 3 $= \sqrt{T_3^2 + (U_3 - S_3)^2} = L_3$

Load on Bearing # 4

Due to $S = \frac{S \times b_2}{c_2} = S - S_3 = S_4$

Due to $T = \dfrac{T \times b_2}{c_2} = T - T_3 = T_4$

Radial load due to thrust $P = \dfrac{P \times r_2}{c_2} = U_3 = U_4$

Total radial load $= \sqrt{T_4^2 + (S_4 + U_4)^2} = R_4$

Thrust load $= P$

Total load on bearing $\# 4 = \sqrt{R_4^2 + P^2} = L_4$

CHAPTER XVI

HORSEPOWER REQUIRED TO DRIVE MOTOR VEHICLES

In Vol. I, Chap. XX, it was shown that the brake horsepower hp. $= \frac{TR}{5{,}252.1}$, where T is the torque in pound-feet, and R the number of revolutions per minute of the engine flywheel, or of the rear wheels, if the hp. of a motor vehicle is determined at the rear wheels. If the torque T is given in pound-inches, hp. $= \frac{TR}{5{,}252.1 \times 12} = \frac{TR}{63{,}025}$, from which $T = \frac{\text{hp.} \times 63{,}025}{R}$, pound-inches, or, $\frac{\text{hp.} \times 5{,}252.1}{R}$ pound-feet.

In Vol. I, Chap. V, it was stated that practice has demonstrated that a maximum torque of about 6¾ lb.-inches is obtained per cubic inch of piston displacement, so that when the required torque at the rear wheels is known, the number of revolutions R of the rear wheels, the reduction and the transmission efficiency, it is possible to determine the engine power or the cylinder dimensions.

Suppose, for instance, that it is desired to find the necessary power to drive a truck, whose total weight with load is 10,000 lb., up a 3-per cent slope, at 20 miles per hour, on the high gear. The average traction resistance of a motor vehicle over ordinary roads (including friction losses in bearings) may be assumed as 50 lb. per ton, or 25 lb. per 1,000 lb. of total weight of truck and load. Hence for a truck whose total weight is 10,000 lb. the tractive effort necessary to drive it on a level road is $\frac{10{,}000 \times 25}{1{,}000} =$ 250 lb. When going up a 3-per cent grade there is an additional tractive effort due to the grade which $= W \times$ the grade. Therefore, the total necessary *tractive effort* when going up a grade is $TE = \left[\frac{W \times 25}{1{,}000} + Wg\right]$, where W is the total weight in pound and g the grade in $\frac{\text{per cent}}{100}$. Hence

$$TE = \frac{10{,}000 \times 25}{1{,}000} + 10{,}000 \times .03 = 550 \text{ lb.}$$

If m is the speed of the car in miles per hour, the speed in feet per minute $= \frac{m \times 5{,}280}{60} = 88\ m$, and the hp. required, to overcome the traction resistance,

$$\text{hp.} = \frac{\text{tractive effort} \times 88m}{33{,}000} = \frac{\left(\frac{W \times 25}{1{,}000} + Wg\right) \times 88m}{33{,}000}$$

If W is 10,000 lb.; m, 20 miles per hour; and g, a 3-per cent grade (*i.e.*, .03) then the necessary

$$\text{hp.} = \frac{\left(\frac{10{,}000 \times 25}{1{,}000} + 10{,}000 \times .03\right) \times 88 \times 20}{33{,}000} = 29.3$$

The transmission efficiency when in the high gear may be assumed as 85 per cent, hence a certain amount of extra power is necessary to compensate for this loss of 15 per cent.

For fast moving vehicles the wind resistance must also be considered as mentioned in Vol. I, Chap. XXV.

The hp. or the torque of the engine may also be determined from the tractive effort. We saw before that the total tractive effort of the truck in question was 550 lb. If the reduction e in the rear axle is 8:1, the efficiency f, 85 per cent, as before mentioned, the torque T of the engine can be found from the equation, $T = \frac{TE \times r}{e \times f}$.

If the radius r of the driving wheels is 18 in. or 1.5 ft., the torque $T = \frac{550 \times 1.5}{8 \times .85} = 121.3$ lb.-feet., or $121.3 \times 12 = 1{,}455$ lb.-inches, or let us say 1,500 lb.-inches. Suppose the reduction in the transmission is 5 : 1, hence the total reduction is $5 \times 8 = 40 : 1$, and the transmission efficiency in the low gear 70 per cent.

When the engine torque is known, the *tractive effort* may be found from the equation $TE = \frac{T \times e \times f}{r}$. At low gear $TE = \frac{1{,}500 \times 40 \times .70}{18} = 2{,}333$ lb. The *tractive factor* $TF = \frac{TE}{W}$, that is to say, the tractive effort divided by the total weight W, which in this case we assumed to be 10,000 lb. Hence at the low gear, $TF = \frac{2{,}333}{10{,}000} = .233$, and at the high gear $TF = \frac{550}{10{,}000} = .055$. The tractive factor is the tractive effort per pound of total weight.

We saw before that the tractive effort on a level road was 25 lb. per 1,000 lb. of weight, therefore, to overcome rolling resistance the tractive factor is $\frac{25}{1,000} = .025$. By deducting .025 from .055 (the tractive factor in high gear) we obtain .03 as the remaining effort, which represents $.03 \times 100 = 3$ per cent, or a hill climbing ability equal to a 3-per cent grade.

By deducting .025 (the tractive factor to overcome the tractive resistance) from the tractive factor found for the low gear, we obtain $.233 - .025 = .208$, which shows that on the low gear the hill climbing ability is 20.8 per cent.

In these examples the traction resistance was assumed as 25 lb. per 1,000 lb. of weight. The traction resistance on good roads is usually considerably smaller, about 15 to 18 lb. per 1,000 lb. of load on the wheels.

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